POWER PLANT ENGINEERING

GENERATION - TRANSMISSION - DISTRIBUTION - UTILIZATION

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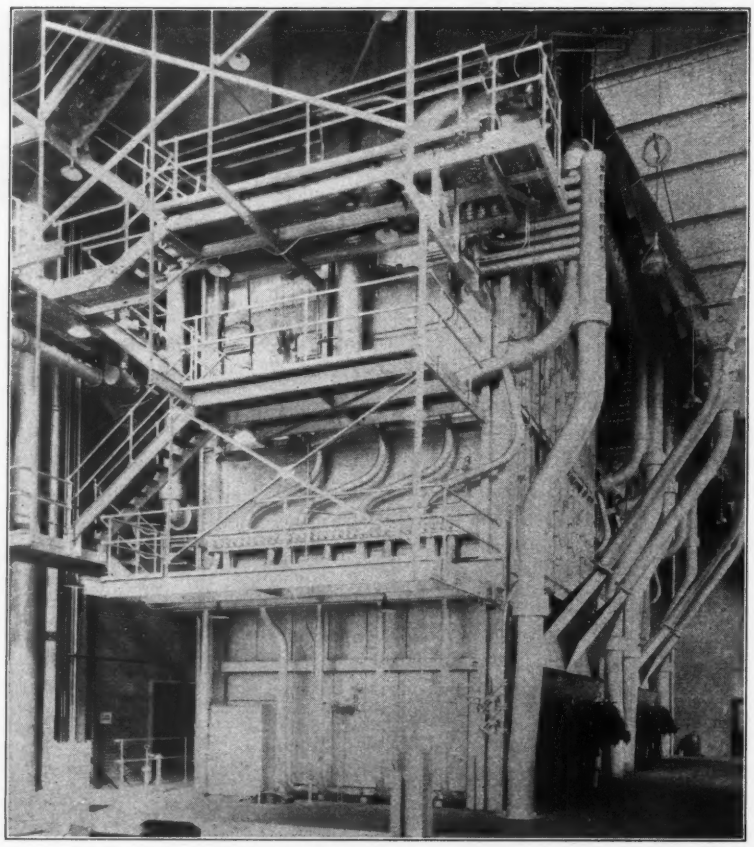
Steam Generators



LEST IT BE THOUGHT indelicate on our part let us hasten to assure you that the illustration at the upper right has nothing to do with liquor. It is not a cross section of a bottle of "Three Star Hennessey" nor does it show what goes on inside of a flask of "Haig and Haig"; rather, it symbolizes a far different kind of noble experiment—one little more successful than the one which terminated last month, but one which in the end served its purpose. For what you see delineated here is the first experimental water tube boiler ever built—the famous three tube, funnel feed, faucet drain, superannuated Blakey model built in 1766. It was not a howling success; we don't know exactly, but something happened to it in the experimental stage and it never went into production. An early document reports "an accident occurring in the inventor's experiment he failed to secure the support he was led to expect."

Despite the accident and the supposed failure of the experiment the idea which it involved led to the development of other "tubular" boilers. But in his wildest dreams Mr. Blakey probably never imagined that this pathetic, funny looking brain child of his would blossom out into the monstrous mass of steel shown in the photograph at the left. For this you will recognize as a modern water tube boiler; not the largest by any means but a highly developed unit, typical of present practice.

It is a far stretch between these two units, not only as concerns steam generators but everything else as well. Nearly all of our intense technical development—practically our entire machine civilization lies between these two boilers. Despite this remarkable progress, development in boiler design and practice is advancing at an amazing rate. In this issue we have endeavored to give you some idea of what has happened during the past ten years. In these pages we present the features of modern systems of steam generation—of pulverized fuel firing, of the development of the water-cooled furnace, of stoker practice. Here we consider how the furnace was turned from a mere chamber for the combustion of fuel, into a complex affair with almost as much heat absorbing surface as the boiler itself and carrying in its bottom a lake of molten slag. Imagine tapping ash from the furnace in liquid form! It was difficult to even imagine this until it happened more or less accidentally at Huntley Station in Buffalo in 1926. Today the slag tap furnace is a successful reality in a score of different stations.

Yet, if these modern developments are astonishing, what will the future boiler look like? We cannot tell, but if some of the more revolutionary developments in Europe may serve as indicators there is still much development to be expected. The Loeffler, the Atmos, the Benson and the Schmitt-Hartman boilers are so unique, so different from anything else we ever thought of in boilers that we cannot believe that our present American units, efficient and reliable though they may be, represent the ultimate in steam generation. The future still has much in store for us.

WITH THE EDITORS

An Established Necessity

AMID ALL THE turmoil of social conditions which is now forcing its attention upon all of us it is well to get off in a corner somewhere with charts and other data we know to be reliable and here formulate a general program to be carried out for a definite period of time, take a look at where we stand, visualize what we wish to accomplish and plan to take what we consider the essential steps to reach the goal. A few of our actions are definitely under our control, a few more may be brought there if proper measures are taken, but the vast majority of the conditions under which we live are not subject to individual control in our complexed society, so to avoid destructive clashes between various classes a common frame of mind must be reached—a willingness to work with others in measures of common interest for which we are not directly personally responsible.

Engineers of power plants as a class have a definite responsibility in our present social scheme. Through them nature's store of fuel may be efficiently or wastefully used, the energy available in running streams may be transformed and transmitted for utilization by the masses of people, much of the waste resulting from industrial processes and even the garbage out of city alleys may be converted into useful energy. They are the lamp lighters of our city streets, office buildings, factories and homes. They are responsible to the community for power to transport people to and from work. In factories a thousand services to workmen depend upon the ability and faithfulness of the power plant engineers.

From the technical point of view there has been continuous, in fact increasing, development throughout the entire period we know as the depression, regardless of curtailed budgets. More efficient, more reliable equipment is available. The same can be said of operating methods. Science has greatly extended the possible services to the public which the power plant can render, be it the utility, industrial, institutional or office building plant.

What disturbing conditions there may be in this field come from outside the power plant itself and its immediate management. No serious difficulties have arisen between employers and employes, working conditions are continuously being improved, no evidence has come to light in recent years where engineers have been unfaithful to their tasks to the extent of sabotage. Outside the plant, however, conditions are not so comfortable. The breaking point between the public and the utilities has been reached in a number of cases. Load requirements in all plants are far too low for economical operation. Government projects are being planned which will still further reduce the uneconomically low plant factor which now prevails. These points of difficulties are in fact evidence of the importance of this great industry and because of its importance the difficulties are sure to be ironed out.

All this leads us to the feeling that the profession of power plant engineering has established for itself a place in the social life of the country that is almost as permanent as the human race itself. It has become nearly as much a necessity of life as food, clothing and shelter. And the part engineers have to play in this great drama is to continue with the improvements they have made and the expansion of the services they can render.

Domestic Load Development

ENCOURAGING increases in domestic consumption of electricity during the depression have led utilities to believe that their future growth lies largely in that direction. During the last year or so this opinion has been well entrenched, the encouraging electric refrigerator and electric range companies of the N.E.L.A. showing what may be expected from a well organized, well financed campaign.

It has, however, been just as conclusively proven that this load cannot be added in any considerable quantity, in the face of rates which are not competitive. Under the conventional and perhaps equitable rate structures prevailing, such competitive rates were out of the question. If each portion of the load is to bear its just share of generation and distribution expense, rates of one and a half to two and a half cents per kilowatt-hour for low and scattered domestic loads were impossible. In the face of rates which average 5.5 cents, building up the load depended largely on salesmanship and resulted only in few scattered cases which affected the situation as a whole not at all.

It is with considerable interest, therefore, that the electrical industry watches the development of the T.V.A. program and policy in this regard. At Tupelo, for instance, the first of the T.V.A. contracts, maximum rates have been specified, the 50 kw-hr. or less load taking a three cent rate. Difficulties of serving this class under this rate are foreseen by a stipulation in the contract that allows the authorities to impose a development surcharge on commercial and industrial business to make up the expected small load deficit.

In addition to taking care of the sale of power to the small user at a loss, a special fund of a million dollars has been made available in the Tennessee valley to finance the purchase of electric appliances. Conditions for testing the validity of claims made for the importance of the domestic load are ideal. Cheap rates and available money for appliances leave little to be desired.

Equally important to the municipalities and public utility companies is the precedent set by such a body as the T.V.A., recognizing the necessity of a subsidy of some kind for business that must temporarily be done at a loss in order to build up a market. The domestic load gain should soon be proven a fact or myth and the practical and economic need for promotional rates for this class definitely determined.

Steam Generating Units Temperature Demands

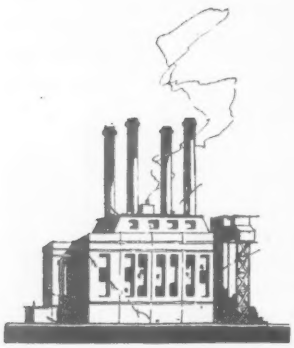
General Consideration of Factors Affecting Distribution of Surfaces in Steam Generating Units. Pressures of 3200 lb. and Temperatures of 1000 deg. Now in Use

DURING the past decade, developments in steam generating units have been directed toward meeting the demands for higher pressure and higher temperature steam and of meeting these demands with increased economy and efficiency. During this time the boiler proper has declined in importance until in the latest units[1] it is little more than an appurtenance serving to screen the superheater elements from slag and from excessive gas temperature.

Need for higher pressure and temperatures has existed since the earliest days of steam power and was fully appreciated by early engineers, some of them contemporary with Watt. It was not until metallurgical developments, steadily increasing electric loads and high coal prices combined after the war to make developments commercially desirable and available, that real progress was made. Then, almost simultaneously, 450, 600, 1200 and 3200 lb. pressures came into commercial or experimental prominence.

1000 Deg. F. Now in Use

Higher temperatures followed somewhat slower, the maximum commercial range rising from 650 to 750, 825 and finally 1000 deg. F. about 1931. In this development Europe led this country somewhat, the method of approach being fundamentally different. Europeans examining the subject theoretically, immediately favored higher temperature with no intermediate steps. Americans more conservatively proceeded first to exhaust the possibilities of higher pressures and then turned to higher temperatures. Today in this country there is a decided feeling that moderate pressures and higher temperatures offer advantage over extreme pressures by eliminating the need for reheat.



Just why these high pressures and temperatures are desirable is shown by Fig. 1. For every pound of steam through the prime mover there is a condenser loss which can be considered approximately constant at 950 B.t.u. If the heat extracted by the prime mover can be raised from 250 to 500 B.t.u. per lb. the thermal efficiency would be increased from $250 \div (250 + 950) = 20.9$ per cent to $500 \div (500 + 950) = 34.5$ per cent. For straight power generation high temperature is more important due to the necessity for keeping moisture at the turbine exhaust within about 10 per cent.[2] Where process steam is needed at definite pressures, more than a few degrees of superheat is usually superfluous. With exhaust conditions fixed in this way, by-product power generation in usable quantities requires higher pressures rather than higher temperatures. This fact, verified by a glance at a Mollier chart,[3] has led to the apparently paradoxical statement that fundamentally central stations need higher temperatures and industrial plants higher pressures.

Development of Boilers Dependent Upon Metallurgical Research

The advantages of going to higher temperatures rather than higher pressures, while obvious enough from a purely thermodynamic standpoint are beset with difficulties when attempts are made to make use of them. The use of still higher temperatures is dependent upon the researches of the metallurgist—upon the development of steel alloys capable of withstanding these higher temperatures and pressures. At the present time we are working very close to the limits of the materials available. At the Detroit Edison Co. at Trenton Channel, a superheater has been operated over 21,000 hr. at temperatures which varied from 1000 to 1100 deg. F., this at a pressure of 400 lb. per sq. in. Thus plants operating at a steam temperature of 1000 deg. seem quite possible.

The material used in these superheaters is an austenitic 18 per cent chromium, 8 per cent nickel alloy. Results of examinations made on various materials from this installation show that some changes have taken place but these are not serious. Loss of strength or tendency toward embrittlement has not progressed to such an extent that consideration has been given to the replacement of any parts.

At the same time boiler designers were striving to meet new conditions they were improving the efficiency and economy of the boiler by increasing its reliability

[1] Buzzard Point Plant Serves Capital. By H. G. Thielscher, Power Plant Engineering, November, 1933, p. 478.
[2] Turbine Performance from the Mollier Diagram, Power Plant Engineering, November 15, 1931, p. 1095.
[3] Turbine Expansion Lines of Actual Machines, Power Plant Engineering, December 1, 1931, p. 1141.

Meet All Pressure *and*

Automatic and simultaneous welding of the girth seams of pressure vessels.

and capacity and by adding heat recovery equipment. Efficiency, capacity and economy (insofar as affected by cost) are closely related. Boiler capacity is determined largely by the limiting efficiency or the temperature to which the gases must be cooled.

In order to cool the gases to a given stack temperature certain heating surface, or combination of surface, is necessary. Experience has shown that in general it makes little difference in the total requirements whether the surface is put in the furnace or in the boiler. If low gas temperature is necessary a large surface, with correspondingly low unit capacity, is necessary; with high gas temperature as when a steaming economizer is used, a small surface with a correspondingly large unit capacity, is allowable. As far as actual heat efficiency is concerned, the early boiler forms, for instance, Scotch Marine and return tubular, are capable of as good performance and modern designs.

It is very natural, therefore, that one ask, "If that is true why have the more complicated designs with water walls, superheaters, economizers and air heaters, been so widely used?" The answer is that the usual simple boiler has size, pressure and efficiency limitations which rule it out of the high pressure field.

The h.r.t. boiler, for instance, is cheap and reliable. It is, however, in usual designs limited to a maximum pressure of about 150 lb. and to maximum sizes about 96 in. in diameter with tubes 20 ft. long giving a surface of about 3500 sq. ft. There is also a limit to the efficiency possible because it is obviously impossible to cool the gases lower than the saturated steam temperature. As a matter of fact the temperature differential is often 100 to 150 deg. F., which for 150 lb. pressure steam would mean a gas temperature of 450 to 500 deg. F. At higher capacities this differential increases giving a typical efficiency curve as shown by Fig. 2.

It is surprising to many to find that the point of maximum efficiency of the small boiler is at 100 per cent rating. This, however, is not mere coincidence. In the early days the amount of steam required by an engine was determined more or less arbitrarily and called a boiler horsepower. Through boiler tests it was then found that it took about 10 sq. ft. of heating surface to produce this quantity of steam at maximum efficiency. The boiler horsepower was therefore arbitrarily placed at 10 sq. ft. of heating surface, meaning that 10 sq. ft. of boiler surface was expected to absorb a boiler horsepower (or 33,479 B.t.u.) when operating at maximum efficiency. This was known as 100 per cent rating, a term still used but having little or no meaning when used with modern steam generating units.

As long as pressures were low and operators satisfied with 100 per cent rating, all was well. Finally, however, combustion equipment was greatly improved and the temptation to cut investment costs by pushing the boilers harder was too great to be resisted. Boiler rating went up and with it exit gas losses until someone had the idea of putting a feedwater heater in the gas stream to reduce the flue gas loss. This developed into the modern economizer, which is much cheaper than boiler surface and, as it can be made countercurrent, the flue gases can be cooled almost to the temperature of the incoming feedwater. The effect of the economizer on boiler capacity and efficiency is shown by Fig. 3.

Stokers and firing methods were improved, more

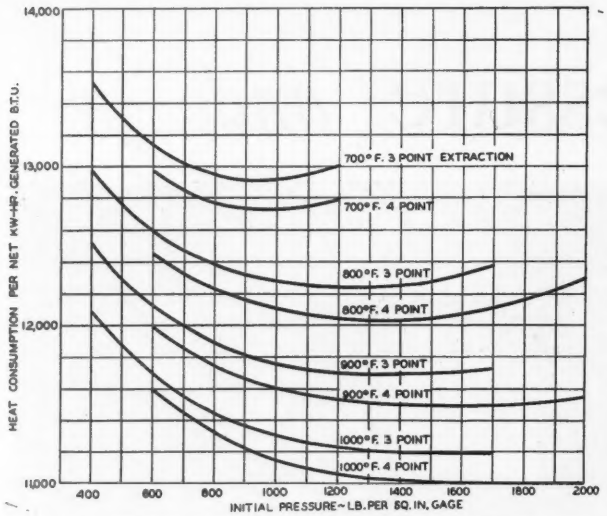


FIG. 1. EFFECT OF PRESSURE AND TEMPERATURE CHANGES ON STATION HEAT CONSUMPTION FOR STRAIGHT REGENERATIVE CYCLE WITH THREE AND FOUR POINT EXTRACTION

coal was burned per square foot of grate surface, forced and reduced draft fans were added to get the necessary air into the furnace and to overcome the additional draft loss due to the economizer. In fact, all went well until the introduction of the regenerative cycle with extraction feedwater heaters. The economizer was out of a job. It was better to heat the feedwater with heat salvaged from the condenser than to save it from the flue gases.

This heat could not be thrown away, however, and the air heater appeared on the scene. Economizers were scrapped, air heaters were the things to use and their effect on capacity and efficiency is shown by Fig. 4. About this time pulverized coal was struggling along with long flame burners and tremendous furnaces. It was discovered that preheated air improved combustion. The turbulent burner was also developed but the high heat releases possible were hard on the refractory walls. Air cooled and later water cooled walls were brought in. Both heated air and water walls were applied to stokers and furnace volumes for both types of firing reached a more staple figure.

Water wall surface was gradually increased to the point where today half the total steam produced by the unit may come from this source. Experience also showed that there was a considerable economic temperature differential between the maximum extraction temperature and saturated drum temperature. Contrary to earlier opinions that economizers did not fit into the picture with feedwater temperatures much in excess of 212 deg. F., it was found that economizers fit into the scheme very nicely when taking water from the high pressure extraction heater and heating it to the steaming point. The steaming economizer was developed and the modern boiler has water walls, superheater, economizer, and air heater. The heat absorption of the various parts in a modern boiler is shown by later articles.

All this equipment costs money. The plant with lots of space, cheap fuel and low pressure can still use the h.r.t. boiler to advantage. A 350-hp. h.r.t. boiler capacity of about 10,000 lb. per hr. of steam will occupy an actual volume of perhaps 6000 cu. ft. or 0.6 cu. ft. per thousand pounds of steam per hour. As space becomes more valuable and fuel more expensive, more elaborate equipment is justified, the average industrial power boiler today having a superheater and economizer with perhaps some water wall surface. Average ratings range from perhaps 200 to 400 per cent and boiler volumes from a third to a half of the above figure.

The whole question of boiler development is one which involves not only the boiler proper—that is that portion of the steam generating unit where water is actually evaporated, but the furnace, firing equipment, economizer and air preheaters as well. As pointed out above, in order to cool the gases to a desired stack temperature a certain combination of heating surfaces are necessary and it makes little difference in the total requirements how this heating surface is distributed. In recent years much more of this surface has been placed in the furnace, and so the furnace becomes much more an integral part of the boiler than it was when it was merely a refractory lined combustion chamber. Water walls have become almost universal for all larger units. Indeed there has been an enormous development in furnace practice and design.

Progress has been made in the design and construction of so-called wet bottom or slag-tap furnaces. Furnace shapes and burner designs more suitable for this type of furnace have been developed. The use of water-cooled furnace floors has greatly reduced the operating difficulties present with this type of furnace. The "dry" versus "wet" furnace, however, is still a moot question, each type having its apparent advantages and disadvantages.

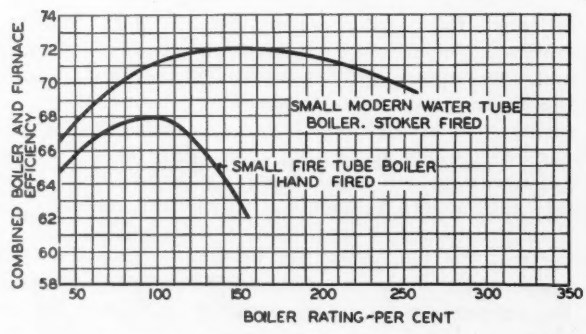


FIG. 2. TYPICAL EFFICIENCY CURVES OF A SIMPLE BOILER

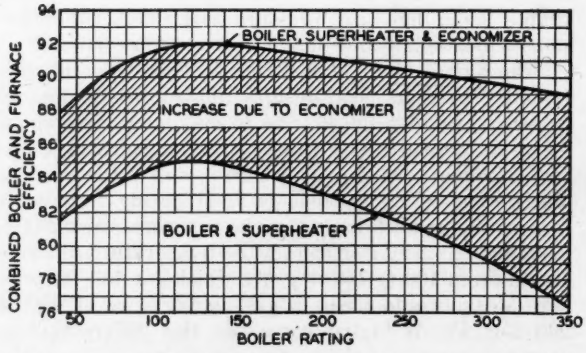


FIG. 3. THE ECONOMIZER INCREASES THE EFFICIENCY AND CAPACITY OF A BOILER

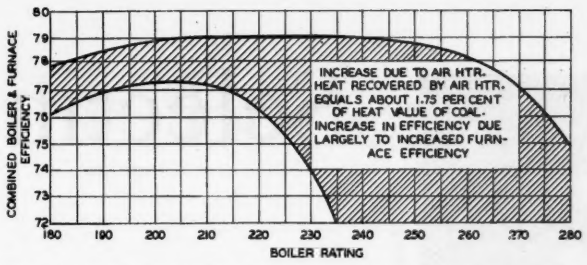


FIG. 4. AIR PREHEATERS ALSO INCREASE THE EFFICIENCY AND CAPACITY OF BOILERS MUCH BEYOND THE ACTUAL HEAT RECOVERED

So too, stokers have undergone great development in the past ten years and have greatly increased in size. This development in stoker practice was induced to a certain extent by the development in pulverized coal practice. Pulverized coal practice led to the use of highly preheated air and this was then incorporated in stoker practice. Today stokers are designed for the use of preheated air as high as 500 and 600 deg.

In pulverized fuel practice methods of firing have changed. Since the original installations of pulverized coal firing, three types of burners have been developed and simplified. Originally all firing was vertically downwards. Later, these burners were designed to admit preheated air which improved the performance. The horizontal burner has been greatly improved and has advantages for certain types of furnaces. The latest type of burner is the corner fired tangential type which fires the coal horizontally from the corners of the furnace and the results obtained with this most modern design have been very successful. This type of firing has been used only on large, nearly square furnaces. Corner firing making use of the kinetic energy of the jets and the differences in velocities of the coal and air, gives the maximum turbulence and scrubbing of the burning coal particles in the hot zone of the furnace rather than depending on mechanical effects such as baffles, vanes, etc., to mix the coal and air before leaving the burner.

Based on results of research, modern air preheaters can be designed for the most efficient plate spacing and arrangement of passages for uniform distribution of the air and gas streams.

The development during the past decade in pulverized fuel firing systems, mechanical stokers, air preheaters, economizers, boilers and furnace construction have resulted in an increase of ten per cent in the efficiencies of similar sized units.

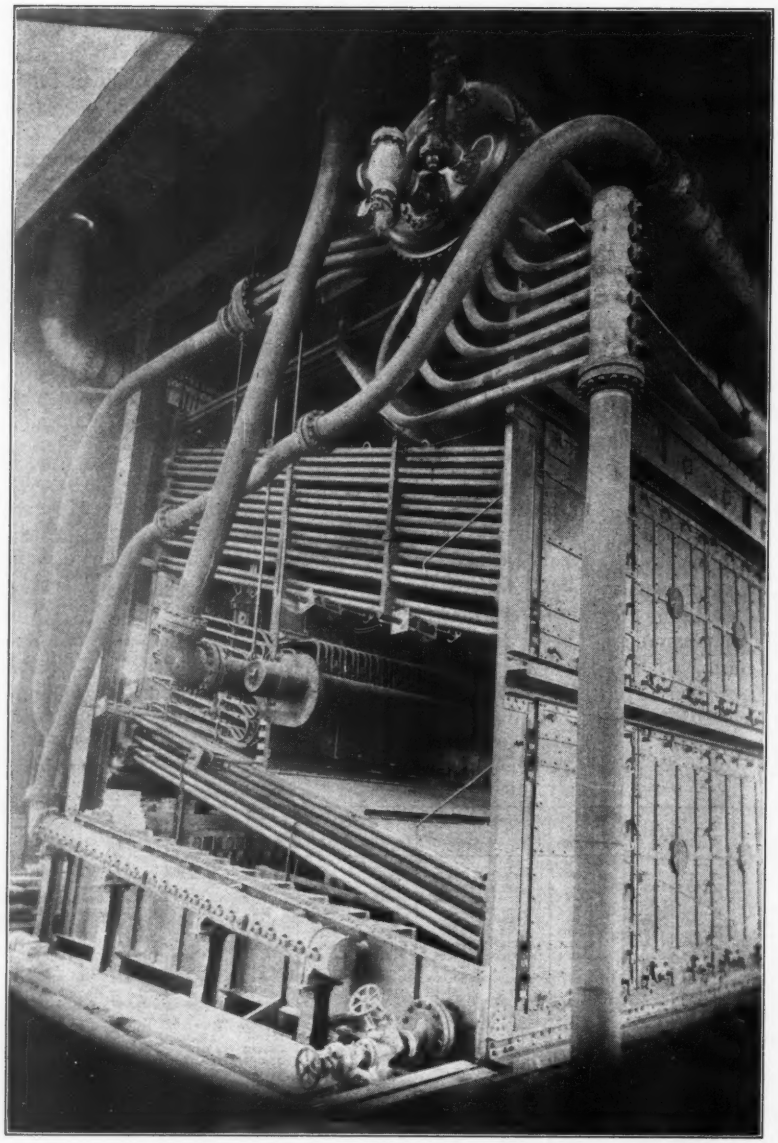
In previous years, boiler room designers usually gave first consideration to the selection of a boiler which when operating at a given per cent of rating would carry the required load. After the boiler dimensions were determined the furnace was fitted to the boiler.

In modern practice, boiler room designers give first consideration to the furnace for all sizes of installations and all types of firing to obtain an economical and efficient design.

COMPLETE DATA ON REPRESENTATIVE CENTRAL STATION AND INDUSTRIAL PLANTS COMPLETED DURING THE PAST FEW YEARS

Name of Station	Capacity Lb/hr. or %	Pressure Lb.Abs.	Temperature Deg.F.	Method of Firing	Boiler H.S. Sq.Ft.	Waterwalls Sq. Ft.	Waterwalls % B.H.S.	Superheater Sq. Ft.	Superheater % B.H.S.	Economizer Sq. Ft.	Economizer % B.H.S.	Air Heater Sq. Ft.	Air Heater % B.H.S.	Furnace Vol. Cu. Ft.	Furnace Vol. Cu.ft./Sq.ft.
Iron Mt.	200%	265	505	Pulv.Coal	13,610	281	2.06	600	5.87	None	-	None	-	10,000	0.735
Dresser	?	365	650	Stoker	17,680	None	-	2,424	11.82	None	-	None	-	4,014	0.232
Nocaho	200%	365	685	Stoker	12,000	None	-	2,008	16.70	3245	27.1	None	-	6,200	0.515
Valmont	?	390	640	Pulv.Coal	10,845	302	2.78	6,288	57.90	2474	22.8	None	-	8,160	0.751
Narragansett	200%	390	675	Pulv.Coal	17,084	1,650	9.67	4,316	25.3	None	-	14,500	85.0	13,000	0.762
Avon	?	405	725	Pulv.Coal	30,600	782	2.59	6,550	22.40	21,180	69.1	None	-	30,600	1.000
Long Beach	300%	410	725	Gas & Oil	14,916	256	1.98	3,574	25.90	6,904	45.5	None	-	5,600	0.375
Richmond	?	415	707	Stoker	15,697	730	3.70	2,822	18.00	7,515	47.9	None	-	7,800	0.496
Columbia	?	615	750	Pulv.Coal	15,110	None	-	2,760	18.25	7,300	48.4	9,868	65.4	12,568	0.830
Trenton Channel	?	430	707	Pulv.Coal	29,097	1,220	4.08	6,070	20.30	18,984	63.5	None	-	25,140	0.841
Massillon	?	315	?	Stokers	8,900	258	26.4	760	8.80	4,600	51.7	?	-	2,375	0.267
Harbor Point	200%	315	540	Stokers	16,100	500	3.10	3,840	23.80	None	-	None	-	6,850	0.425
Bayside	150,000	350	700	Pulv.Coal	13,010	1,500	11.5	4,100	31.50	7,390	56.6	?	-	10,700	0.825
Neches	200%	390	700	Oil & Gas	15,300	1,440	9.42	3,718	24.35	None	-	Future	-	8,050	0.525
E. River	?	390	700	Pulv.Coal	14,809	4,134	27.7	3,430	23.00	None	-	25,900	194.0	20,857	1.405
Des Moines	250%	400	695	Stoker	15,018	310	2.06	2,900	19.30	1000	6.6	12,420	82.7	6,183	0.410
Lauderdale	?	465	700	Oil	17,440	3,187	18.00	4,157	23.80	None	-	82,000	470.0	15,600	0.895
Lakeside	250%	1405	735	Pulv.Coal	22,528	1,284	5.39	951.5	3.27	None	-	67,500	254.0	30,100	1.086
Hell Gate	720%	315	735	Pulv.Coal	23,800	3,800	15.95	4,800	17.15	26,136	109.3	18,840	79.3	18,000	0.756
Cahokia	400%	350	725	Pulv.Coal	18,010	1,675	7.90	3,789	21.0	None	-	9,920	55.0	12,000	0.666
Trinidad	272%	405	750	Pulv.coal	19,471	1,805	5.56	4,000	20.5	None	-	None	-	18,540	0.953
Stanolind	400%	415-155	650-475	Pulv.coal	17,546	2,250	12.80	3365-1682	19.2-9.6	6,136	46.3	10,564	60.1	15,680	0.892
Saginaw River	350%	415	725	Stoker	12,540	749	6.06	5,856	43.8	None	-	12,000	97.2	6,000	0.485
Borg Paper	455%	430	725	Pulv.Coal	5,790	530	9.15	1,548	26.7	2,969	51.2	7,350	127.0	4,300	0.743
Long Beach #3	322%	460	750	Gas & Oil	34,182	3,240	9.5	6,553	19.2	None	-	51,232	150.0	19,800	0.580
Hudson Ave.	500%	465	725	Stokers	23,764	1,326	5.59	3,037	12.8	14,975	63.0	?	-	8,500	0.358
Edgar	300%	1415	735	Stokers	16,095	1,235	6.2	3,485	21.65	5,595	35.0	33,052	209.0	9,996	0.626
Ohio River	140,000	?	690	Stokers	11,520	820	6.98	2,150	18.65	8,640	75.0	None	-	5,800	0.484
Fremont	54,000	335	550	Stokers	6,080	None	-	1,782	29.3	864	14.2	2,478	40.7	1,620	0.265
Cahokia	200,000	350	750	Pulv.Coal	18,010	1,835	10.19	4,070	22.6	None	-	11,700	65.0	14,650	0.814
E.Riv. Addition	800,000	390	700	Pulv.Coal	60,014	7,345	12.20	13,900	23.1	-	-	91,758	15.3	38,200	0.636
Delray No. 3	340,000	409	725	Stoker	23,978	3,802	15.85	4,900	20.4	6,032	25.1	29,790	124.1	13,800	0.577
Miami Copper Co.	65,000	435	650	Oil	7,200	396	5.50	1,085	15.08	None	-	7,950	110.4	4,200	0.583
Gorgas #2	500,000	475	750	Pulv.Coal	30,500	3,505	11.56	4,991	16.45	None	-	30,600	101.0	24,000	0.791
State Line	450,000	650	750	Pulv.Coal	10,134	4,441	43.10	7,800	75.6	19,820	192.5	66,401	647.0	22,700	2.180
Station A	500,000	1400	750	Gas & Oil	10,699	1,617	15.1	7,430	69.5	20,944	196.0	48,915	459.0	15,100	1.410
Holland	300,000	1415	750	Pulv.Coal	7,928	2,194	27.6	5,500	69.5	11,880	150.0	32,450	410.0	16,500	2.080
Atwater Kent	85,000	215	486	Pulv.Coal	7,203	4,124	56.2	950	13.18	-	-	11,250	156.0	5,750	0.797
Belle Isle	180,000	400	735	Pulv.Coal	12,670	648	5.12	606	4.79	-	-	22,000	174.6	6,700	0.530
Atkinson	450,000	440	735	Pulv.Coal	30,250	3,580	11.85	4,590	14.85	-	-	30,000	99.4	25,120	0.834
Procter & Gamble	60,000	465	750	Stoker	5,531	198	3.58	710	12.85	-	-	4,554	83.9	3,320	0.480
Rio Grande	184,500	465	750	Gas & Oil	15,255	1,249	8.20	4,785	31.4	-	-	19,450	127.5	9,200	0.603
Champion Coated Paper	345,000	670	750	Pulv.Coal	27,192	2,620	9.65	5,300	19.5	-	-	47,742	175.5	19,080	0.701
Louisiana	350,000	635	750	Oil & Gas	15,817	1,867	13.45	7,665	55.5	9,801	70.9	32,800	237.0	16,600	1.210
South Amboy	275,000	1400	750	Pulv.Coal	7,214	2,036	28.15	12,900	178.7	4,425	61.2	32,082	444.0	12,900	1.790
Deepwater	350,000	1450	830	Oil & Gas	7,778	1,473	18.95	8,200	105.3	13,820	178.0	31,154	400.0	14,000	1.800
Concepcion Road	215,000	1464	810	Oil & Gas	4,990	1,011	20.5	4,640	92.9	11,880	238.0	13,900	279.0	7,018	0.141
M. J. Whittall, Assoc.	50,000	215	486	Stokers	5,063	72	1.42	1,000	19.75	1,093	21.6	-	-	1,500	0.290
F. H. Glatfelter	134,000	365	560	Pulv.Coal	11,380	290	2.55	875	7.7	13,160	115.7	-	-	6,500	0.572
Michigan Limestone & Chemical	80,000	415	700	Stokers	8,797	145	1.65	1,530	7.39	-	-	4,555	51.7	3,000	0.341
Raritan River	190,000	455	725	Stokers	11,810	607	5.15	2,540	21.5	3,840	32.5	17,900	150.7	6,500	0.550
Hudson Avenue	530,000	455	750	Stokers	24,450	3,844	15.71	5,740	23.45	23,400	98.4	-	-	14,000	0.572
Trinidad	385,000	465	750	Pulv.Coal	32,047	5,468	17.07	6,475	20.2	-	-	-	-	19,500	0.608
Glenwood	300,000	465	800	Pulv.Coal	20,036	2,800	13.97	5,000	24.96	-	-	44,325	224.0	11,700	0.584
Moores Park	240,000	585	700	Stokers	16,500	830	5.02	3,600	21.8	7,900	47.9	17,500	106.0	10,000	0.606
Michigan City	345,000	695	770	Pulv.Coal	7,197	2,250	31.3	5,072	70.5	14,929	207.0	48,000	584.0	14,700	2.045
Lakeside	300,000	1405	750	Pulv.Coal	23,640	5,205	22.0	1,125	4.76	-	-	77,500	328.5	28,573	1.210
Purdue	44,740	225	560	Stokers	6,448	279.5	4.34	960	14.9	None	-	-	-	2,687	0.417
Kidder	98,000	325	600	Stokers	7,500	-	-	1,240	16.54	-	-	6,960	92.8	2,200	0.293
Springwell	90,000	395	710	Stokers	10,328	510	4.91	1,720	16.6	2,436	23.4	9,980	96.2	4,128	0.398
Pennsylvania Sugar	300,000	415	585	Pulv.Coal	11,000	6,550	51.5	435	3.96	5,000	45.5	18,750	170.5	19,485	0.948
Pasadena	210,000	415	750	Gas & Oil	15,186	946	6.24	3,340	22.0	None	-	37,000	244.0	7,600	0.500
Quindara	180,000	425	700	Pulv.Coal	16,267	1,955	12.0	5,704	35.1	1,843	11.3	19,500	120.0	10,900	0.670
Dodge City	72,000	440	750	Oil & Gas	6,000	-	-	922	15.35	-	-	9,880	164.1	3,545	0.925
Harding St.	190,000	455	725	Stoker	11,810	833	7.03	2,540	21.4	5,760	48.6	17,800	150.9	8,700	0.845

Selecting



USUALLY a steam generating unit is selected to meet definite and predetermined conditions of load, temperature and pressure, although these conditions should not be set arbitrarily without considering the boiler plant which represents perhaps one-third of the total cost of a power plant. Quite naturally, the cost of equipment goes up with the pressure although not always in the same proportion.[1] Furthermore, some pressures are more popular than others and the quantity production in these ranges brings the cost down. In the immediate past 400 lb. has been a popular industrial pressure and 1350 lb. a popular central station pressure.

In the interest of economy, power plant equipment is built in a number of standardized ranges or series[2] which are 25, 125, 150, 250, 400, 600, 900 and 1350 lb. w.s.p. As piping, fittings and in general all standard equipment conforms to these pressure ranges, it is well to work up to the practical limit when determining operation pressures.

As an indication of economic considerations which influence boiler proportions, a few figures[3] are given to show the effects of various controllable factors in boiler design. These figures are of course not applicable for general use but the relative values indicate the importance of a balanced design. For a 1200 lb., 750 deg. F. plant with the turbine exhausting at 400 lb., each pound pressure loss costs $136 and each degree Fahrenheit temperature drop costs $400 per yr. per 100,000 lb. per hr. rate of flow. For 400 lb., 750 deg. F. initial conditions and 1 in. abs. condenser pressure, the corresponding figures are $50 and $140 with the following additional items: Exit gas temperature $65 per yr. per deg. F. change; induced draft loss $310 per yr. per inch draft loss; boiler efficiency $2300 per yr. for each per cent change in efficiency.

The way in which different designers have met local conditions with well balanced designs are indicated in the table in the preceding article, the data for which were taken from plants covered in the annual review issues of POWER PLANT ENGINEERING for the years 1927-33 inclusive. Details of individual stations may be found in the appropriate review issue or in more detail in articles which appeared during the year.

COAL INFLUENCES DESIGN

It is of course impossible to divorce one item from the other when considering plant design. In designing a plant to meet given steam requirements, it is well to begin with the available coals and work from this point through the furnace to the boiler.[4]

Having decided how the coal available may be

[1] Cost of Plants on a pressure basis, Power Plant Engineering, Nov. 1, 1929, p. 1158

[2] See American Standards Association publications and piping specifications.

[3] N.E.L.A. Proceedings 1931, p. 749.

[4] Selecting Boiler Plant Equipment Intelligently, by John Van Brunt, Power Plant Engineering, July 15, 1931, p. 743.

the Boiler to Meet Definite Needs

burned to best advantage, the type of furnace and range of combustion rates should be fixed. In the burning of pulverized coal in wholly or partly water cooled furnaces, the relations of combustion rate, volume of furnace area of water-cooled walls, combustion air temperature, melting temperature of ash, location of boiler tubes in relation to burners and spacing of boiler tubes, to boiler availability or hours of continuous operation are little understood or appreciated by the average purchaser of boiler plant equipment.

IN THE selection of boilers it is well to work up to the economic limit of the particular class into which the boiler decided upon falls. For instance for a 1200 lb. 750 deg. F. plant with the turbines exhausting at 400 lb. each pound pressure less cost $136 and each degree Fahrenheit temperature drop costs $400 per yr. per 100,000 lb. per hr. rate of flow. ¶For 400 lb. 750 deg. F. and 1 in. abs. condenser pressure the corresponding figures are $50 and $140 with the following additional items: Exit gas temperature $65 per yr. per deg. F. change, induced draft loss $310 per yr. per in. draft loss; boiler efficiency $2300 per yr. for each per cent change in efficiency.

High combustion rates of low or medium fusion temperature ash coal results in rapid destruction of furnace walls. Low combustion rates with good coal are permissible in refractory furnaces.

Average coal having an ash fusion temperature of 2100 to 2500 deg. should be burned in water-cooled furnaces and the combustion rate should be so proportioned that the temperature of gas entering the first bank of boiler tubes will be lower than the ash fusion temperature for continuous operation. For short swings, the temperature may be higher, depending on the duration of the swings.

For example, while it may be possible to operate at 25,000 B.t.u. per cu. ft. for one hour and 23,000 for three hours, any attempt to go beyond 18,000 B.t.u. for continuous operation twenty-four hours a day for thirty days would result in fouling of the gas passes of the boiler and building up of clinker in the bottom of the furnace.

A few years ago advocates of high heat release furnaces predicted rates of 50,000 to 100,000 B.t.u. per cu. ft. True, coal may be nearly completely burned at such rates provided it is very finely pulverized. Today, however, experience has demonstrated that such rates are impractical. Excessive slagging of boiler tubes, combustible losses and high cost of pulverizing have placed fairly definite limits on practical combustion rates.

The value of continuous service or availability should be weighed. The desired reliability may be obtained by designing for continuous service or by providing spare capacity in an additional boiler so that a regular schedule of outage may be established for all necessary cleaning and repairs.

Knowing the kind of coal to be used, the amount, pressure and temperature of steam and having determined the method of firing, the combustion rate and size of furnace, the engineer is now in a position to proceed with the selection of the boiler and the number of units required.

If the boiler must fit into an existing boiler room or between existing columns and beneath existing girders, the choice is narrowed as is the case if the space available is otherwise limited. If a new building is contemplated, the boiler may be as large as may seem desirable and the number of units may be reduced to a minimum.

With the number of units known and the furnace volume determined from the combustion rate, an approximation of the furnace width may follow. A very narrow boiler may not be suitable as in order to obtain the required volume in furnace, the height of furnace or depth from front to back or both may be too great to obtain a correctly shaped furnace. For example, if 10,000 cu. ft. are required, a boiler 20 ft. wide with furnace 25 ft. high and 20 ft. deep would be well proportioned while a furnace 12 ft. wide would be 30 ft. high and 28 ft. deep, absurdly ill proportioned. Also, in a boiler and furnace too narrow, the draft loss will be too high.

Factors determining the selection are: first cost; space occupied; space necessary for renewal of tubes; space in which boiler must be located; whether economizers or air preheaters are to be used; space occupied by economizers or air preheaters.

Frequently, it is desired to extend an existing boiler room to utilize existing coal handling apparatus. The space then available may be a deciding factor in the type of boiler. For example, extending a coal bunker may so block the aisle in front of the boiler that there will be no room to remove tubes from a sectional header boiler. It may not be possible to remove them from the rear, particularly if an economizer or air heater must be placed in the rear. A bent tube boiler will often meet that condition and in

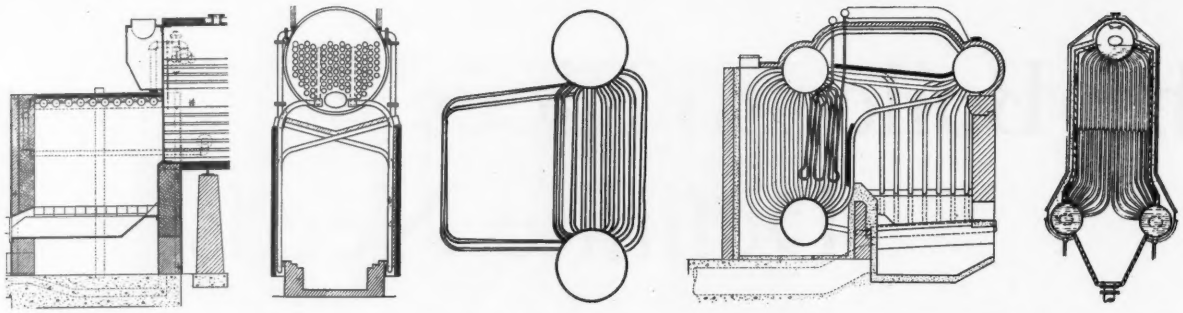


FIG. 1. A. A TWO PASS ADAPTATION OF THE H.R.T. BOILER WITH A DUTCH OVEN. B. WATER WALLS APPLIED TO AN H.R.T. BOILER. C. A SPECIAL DESIGN TO FIT INTO THE SAME SPACE AS AN H.R.T. BOILER. D. ANOTHER APPLICATION OF WATER WALLS TO A LOW HEAD ROOM BOILER. E. A THREE DRUM WASTE HEAT BOILER

addition the space between the inclined tubes and the front wall will add materially to the volume of the furnace.

Boilers of either type operated at over 100,000 B.t.u. per square foot of heating surface will have exit gas temperature 200 to 250 deg. F. above saturated steam temperatures. Hence fuel economy will dictate the installation of an economizer and/or air heater following the boilers. For exit gas temperatures up to 700 deg. or 750 deg. an air preheater will reduce the gas to about 350 deg. and preheat combustion air to 550 or 600 deg. For stoker fired units, this temperature is too high, therefore, the air heater should be preceded by an economizer or the economizer only should be used. Also, in direct fired pulverized coal firing, an air temperature of 500 to 600 deg. F. is under most conditions higher than desirable.

On direct fired units, an air temperature of 300 to 350 deg. will be found sufficiently high for mill drying. Higher temperature than this will necessitate excessive tempering. With an air preheater it is possible to bring the flue gas down to 200 deg. but with few exceptions such a low temperature is not desirable as condensation may occur at temperatures of 300 deg. F. or even higher.

To summarize: first determine the coal or coals to be used, then the method of firing, the combustion rate, the grate surface or furnace volume, next the furnace width and size of boiler, the rate of driving, the type of boiler and the nature of the extended surface, that is, economizer or air heater.

Studies of two or more equally satisfactory arrangements will usually be sufficient to determine the most economical, if at the same time proper weight is given to accessibility for operators and repairs and labor required for routine operation. Only by intimate knowledge of all of the factors and by the exercise of a sound and well-balanced judgment will a correct solution be reached.

Details of firing, furnaces and water walls will be covered in subsequent articles, this one will deal primarily with the boiler types available, the boiler in this case being taken as the saturated steam making surface minus the water walls. With the introduction of extraction heaters steaming economizers and superheaters, the boiler is left to supply only the latent head to the steam. Even this function is encroached upon by the water wall which in large modern boilers may supply as much saturated steam as the boilers.

Field of Various Types Overlap

That, of course, concerns only those interested in large, high pressure installations which, although representing the highest point of the designer's art, are in the minority. There are still hundreds and hundreds of small h.r.t., locomotive type, low head room, bent tube and straight tube boilers sold annually, although many of them, used only for heating, are without even

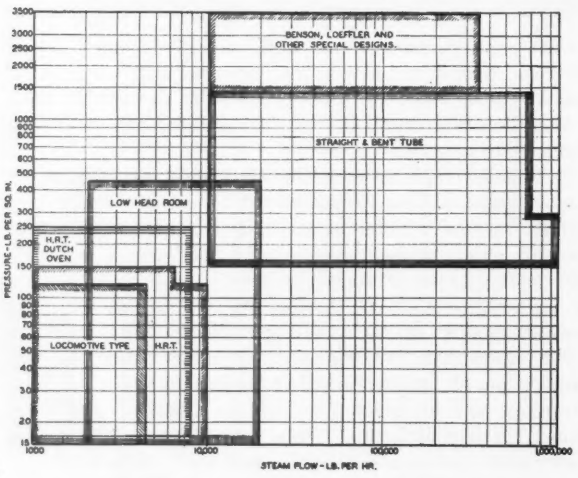


FIG. 2. THE FIELD OF BOILER APPLICATION IS WELL COVERED BY A MULTITUDE OF DESIGNS AND IT IS IMPOSSIBLE TO SET DEFINITE LIMITATIONS ON EACH DESIGN

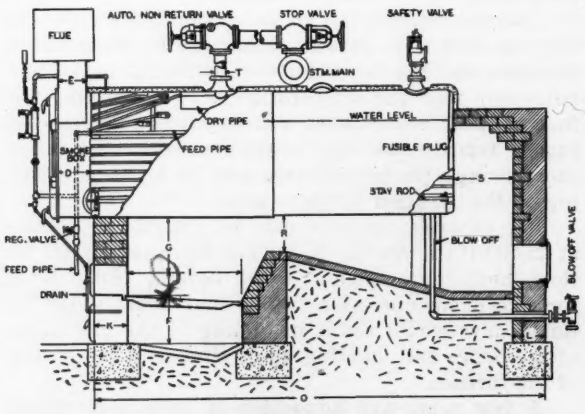


FIG. 3. DETAILS OF AN H.R.T. BOILER

TABLE I. TYPICAL HORSEPOWER RATING AND CONSTRUCTIONAL DETAILS OF MISCELLANEOUS H.R.T. BOILERS

Horse power based on 10 sq.ft. of W.H.S.	Square feet water heating surface	TUBES				Horse power for each additional foot length of tubes	Size of steam nozzle	Size of safety valve nozzle	Size of safety valve required by A.S.M.E. Code for 150 lb. W.P.	Size of feed pipe	Size of blow-off pipe	Size of water column	Flue area required for each boiler	Diameter of stack required for each boiler	Height of stack above grates	FOR GOOD GRADE BITUMINOUS COAL			
		Diameter	Length	Number	Diameter	Length										Width of grate	Length of grate	Grate surface	Ratio grate surface to heating surface
		In.	Ft. In.		In.	Ft.		In.	No. In.	In.	In.	In.	Sq. Ft.	In.	Ft.	In.	In.	Sq. Ft.	
20	198	36	8 11	32	2½	8	2.38	4	4	1 1½	1	1½	1.5	18	50	36	30	7½	26½
25	245	36	10 11	26	3	10	2.36	4	4	1 1½	1	1½	1.5	18	50	36	30	7½	32½
35	352	42	11 0	48	2½	10	3.42	4	4	1 2	1	1½	2.2	20	60	42	36	10½	33½
40	404	42	14 0	34	3	13	3.03	4	4	1 2	1	1½	2.4	22	60	42	42	12¼	33
53	526	48	13 2	50	3	12	4.28	4	4	2 1½	1	1½	2.7	22	60	48	42	14	37½
58	582	48	16 3	44	3	15	3.90	4	4	2 2	1	1½	3.0	24	60	48	48	16	36¼
78	777	54	16 3	60	3	15	5.18	4	4	2 2	1½	1½	3.8	27	70	54	54	20¼	38½
88	883	60	17 5	54	3½	16	5.38	4	4	2 2	1½	1½	4.3	30	70	60	48	20	44½
99	985	60	17 5	72	3	16	6.03	4	4	2 2	1½	1½	4.7	30	70	60	54	22½	43
117	1172	66	17 6	74	3½	16	7.17	6	6	2 3	1½	1½	5.3	33	70	66	60	27½	42½
130	1304	66	17 6	98	3	16	8.00	6	6	2 3	1½	1½	5.7	33	70	66	60	27½	47
146	1463	66	19 6	98	3	18	8.00	6	6	2 3	1½	1½	6.2	36	80	66	66	30¼	48½
156	1562	72	19 7	78	4	18	8.53	6	6	2 3	1½	2	6.5	36	80	72	66	33	47½
167	1674	72	19 7	96	3½	18	9.15	6	6	2 3	1½	2	6.8	36	80	72	66	33	50
183	1827	72	19 7	124	3	18	10.00	6	6	2 3½	1½	2	7.3	39	80	72	72	36	50½
190	1902	78	19 8	96	4	18	10.37	6	6	2 3½	2	2	7.5	39	80	78	72	39	48¾
197	1970	78	19 8	114	3½	18	10.76	6	6	2 3½	2	2	7.7	39	80	78	72	39	50½
211	2107	78	19 8	144	3	18	11.50	6	6	2 3½	2	2½	8.0	39	80	78	78	42¼	50
224	2243	84	19 10	114	4	18	12.25	6	6	2 3½	2	2½	8.5	42	80	84	72	42	53½
243	2425	84	19 10	142	3½	18	13.25	6	6	2 4	2	2½	9.0	42	80	84	78	45½	53½
260	2598	84	19 10	180	3	18	14.2	6	6	2 4	2	2½	9.5	42	80	84	84	49	53
269	2690	84	21 10	142	3½	20	13.25	6	6	2 4	2	2½	9.7	42	80	84	84	49	54½
285	2848	90	19 11	168	3½	18	15.05	6	6	2 4	2	2½	10.2	45	90	90	84	52½	54½
300	3008	90	21 11	140	4	20	14.83	6	6	2 4½	2	2½	10.5	45	90	90	84	52½	51
301	3014	90	19 11	210	3	18	16.48	6	6	2 4½	2	2½	10.5	45	90	90	90	56¼	53½
338	3384	96	22 0	158	4	20	16.66	6	6	2 4½	2	2½	11.8	48	90	96	96	64	53
353	3527	96	20 0	248	3	18	19.32	6	6	2 4½	2	2½	12.3	48	90	96	96	64	55
355	3552	96	22 0	190	3½	20	17.51	6	6	2 4½	2	2½	12.4	48	90	96	96	64	55½

TABLE II. SETTING DATA FOR H.R.T. BOILERS WITH FLUSH FRONTS. LETTERS FOR DIMENSIONS SHOWN ON FIG. 3

Diameter of boiler	Length of boiler over all	Length of tubes	Length of curtain sheet	Size of smoke opening	W Floor line to top of grates	B Top of grate to under side of boiler	Width of furnace	I Length of furnace	Thickness of side wall at bottom	K Thickness of side wall at top	L Thickness of front wall	Thickness of rear wall	Width of combustion chamber	Width of side wall at rear	O Length of boiler setting	Width of boiler setting	Top of side walls from floor line	Top of bridge wall to underside boiler	S Space between boiler and wall	T Diameter of nozzles	Size of blow off	Center to center of boilers in battery	Thick. of center wall at bottom	Common brick required above floor line, single boiler	Number of fire brick required	Additional common brick per foot of boiler length	Additional fire brick per foot of boiler length	Common brick for each boiler set in battery after first	Additional common brick, each boiler set in battery after first per foot of length	Lb. Approximate weight bare boiler	Lb. Additional weight of bare boiler for each foot of tube length	Lb. Approximate weight boiler complete with casings and trimmings
In.	Ft.	Ft.	D In.	In. x In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	Ft. In.	Ft. In.	Ft. In.	In.	In.	In.	In.	Ft. In.	In.									
24	8	8	9	7 x22	24	22	24	30	12	12	15	15	44	15	11- 8	6-6	9- 5	14	12	4	2	4-10	22	5,900	1300	350	75	3,800	175	3,100	350	5,800
42	12	12	11	9x30	24	24	36	36	12	12	15	15	54	15	13- 0	7-6	9- 7	15	12	4	2	5- 4	24	9,000	2000	425	80	4,800	175	4,200	400	7,800
54	15	15	12	10x42	24	32	54	54	12	12	15	15	60	15	14- 0	8-0	9- 8	15	12	4	2	5- 6	24	10,900	2400	450	95	5,900	190	5,500	525	10,500
60	16	16	12	11x46	24	34	60	60	12	12	15	15	66	15	14- 3	8-6	9-10	15	12	4	2	5- 8	24	12,100	2800	475	100	7,800	250	15,200	650	17,200
66	16	16	12	12x46	24	36	66	66	12	12	15	15	72	15	14- 6	8-6	9-11	15	12	4	2	5-10	24	14,100	3200	500	110	9,700	250	18,700	750	20,700
72	18	18	15	13x52	25	42	72	66	12	12	15	15	78	15	22- 7	9-6	9- 7	15	24	4	2½	6- 0	24	16,300	3700	525	120	10,400	280	20,700	1000	27,700
78	18	18	15	14 x66	25	44	78	72	12	12	15	15	84	15	22- 7	10-0	10- 1	15	24	4	2½	6- 2	24	17,300	4000	550	125	11,000	290	23,800	1170	32,300
84	18	18	15	15x66	27	46	84	78	12	12	15	15	90	15	22-10	10-6	10- 3	15	24	4	2½	6- 4	24	18,700	4300	575	135	12,000	300	26,600	1400	36,200
90	20	18	17	17x78	27	48	90	84	12	12	15	15	96	15	23- 0	11-0	11- 5	15	24	4	2½	6- 6	24	21,300	4800	600	140	13,800	310	30,400	1650	47,400
96	20	19	19	19x82	27	48	96	96	12	12	15	15	102	15	23- 0	11-6	11- 6	15	24	4	2½	6- 8	24	22,000	5000	625	145	14,400	320	43,400	1900	55,000

TABLE III. DETAILS OF LOW HEAD ROOM BOILERS AND SETTINGS OF THE DESIGN OR TYPE SHOWN BY FIG. 4A

Heating Surface Sq.Ft.	Tubes Wide	Depth of Setting A	Width of Setting		Width of Furnace	Center Line of Boiler to Battery Wall Centerline	E	F	G	H	J	Width Flue	L	M	Size of Steam Outlet	Size of Feed	No. of Fire & Ash Doors
			Single Boiler	Two Boilers in Battery													
		Ft. In.	Ft.	Ft.	Ft.	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft.	Ft. In.	Ft. In.	In.	Ft. In.	In.	In.	
714	6	15 2	6	11	3	2 6	9 6	7 0	5 2	6	14 5½	2 0	23	14 2½	5	2	1
952	8	15 2	7	13	4	2 6	9 6	7 0	5 2	6	14 5½	2 8	23	14 2½	5	2	1
1190	10	15 2	8	15	5	3 6	9 6	7 0	5 2	6	14 5½	3 4	23	14 2½	5	2	1
1428	12	15 2	9	17	6	4 6	9 6	7 0	5 2	6	14 5½	4 0	23	14 2½	5	2	2
1666	14	15 2	10	19	7	4 6	9 6	7 0	5 2	6	14 5½	4 8	23	14 2½	5	2	2
1904	16	15 2	11	21	8	5 6	9 6	7 0	5 2	6	14 5½	5 4	23	14 2½	5	2	2
2142	18	15 2	12	23	9	5 6	9 6	7 0	5 2	6	14 5½	6 0	23	14 2½	5	2	3
2380	20	15 2	13	25	10	6 6	9 6	7 0	5 2	6	14 5½	6 8	23	14 2½	5	2	3
900	6	17 8	6	11	3	2 6	12 0	7 4½	4 9½	7	14 5½	2 0	23	14 2½	5	2	1
1800	12	17 8	9	17	6	4 6	12 0	7 4½	4 9½	7	14 5½	4 0	23	14 2½	5	2	2
2700	18	17 8	12	23	9	5 6	12 0	7 4½	4 9½	7	14 5½	6 0	23	14 2½	5	2	3
3000	20	17 8	13	25	10	6 6	12 0	7 4½	4 9½	7	14 5½	6 8	23	14 2½	5	2	3
1086	6	20 2	6	11	3	2 6	14 6	7 8½	4 5½	8	14 5½	2 0	23	14 2½	5	2	1
2172	12	20 2	9	17	6	4 6	14 6	7 8½	4 5½	8	14 5½	4 0	23	14 2½	5	2	2
3258	18	20 2	12	23	9	5 6	14 6	7 8½	4 5½	8	14 5½	6 0	23	14 2½	5	2	3
3620	20	20 2	13	25	10	6 6	14 6	7 8½	4 5½	8	14 5½	6 8	23	14 2½	5	2	3
3982	22	20 2	14	27	11	6 6	14 6	7 8½	4 5½	8	14 5½	7 4	23	14 2½	6	2½	3
4344	24	20 2	15	29	12	7 6	14 6	7 8½	4 5½	8	14 5½	8 0	23	14 2½	6	2½	3
1690	8	22 8	7	13	4	3 0	17 0	8 3½	4 1½	8	14 11	4 0	26	14 5½	5	2	1
2113	10	22 8	8	15	5	3 6	17 0	8 3½	4 1½	8	14 11	5 0	26	14 5½	5	2	1
2958	14	22 8	10	19	7	4 6	17 0	8 3½	4 1½	8	14 11	7 0	26	14 5½	5	2	2
3381	16	22 8	11	21	8	5 0	17 0	8 3½	4 1½	8	14 11	8 0	26	14 5½	5	2	2
4226	20	22 8	13	25	10	6 0	17 0	8 3½	4 1½	8	14 11	10 0	26	14 5½	5	2	3
5071	24	22 8	15	29	12	7 0	17 0	8 3½	4 1½	8	14 11	12 0	26	14 5½	6	2½	3

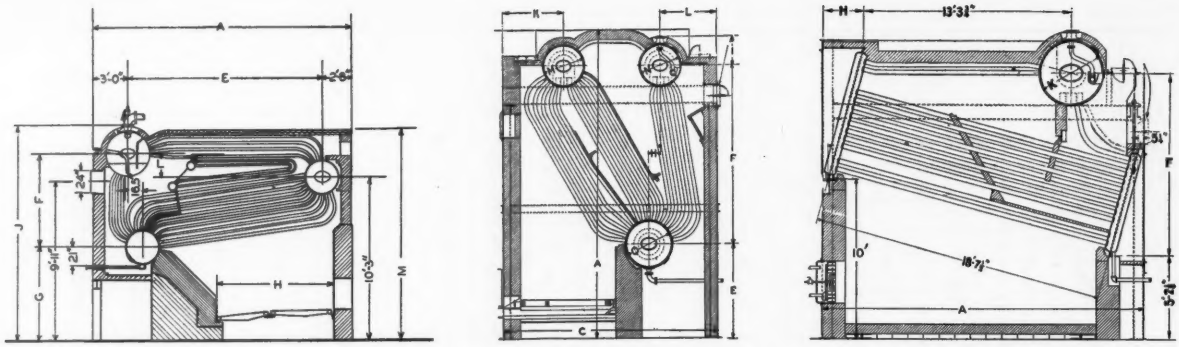


FIG. 4. (LEFT) TYPICAL LOW HEAD ROOM BOILER TO GO IN SAME SPACE AS AN H.R.T. TO BE USED WITH TABLE III. (CENTER) TYPICAL BENT TUBE BOILER DESIGN TO BE USED WITH TABLE IV. (RIGHT) CROSS DRUM STRAIGHT TUBE BOILER TO BE USED WITH TABLE VI

TABLE IV. TYPICAL DATA FOR A NUMBER OF SIZES OF THREE DRUM BENT TUBE BOILERS AS SHOWN BY FIG. 4B

Tubes Wide	Heating Surface	A	Width of Setting	C	Width of Furnace	E	F	H	I	K	L	N	W	O	Thickness of Walls			Steam Outlet	Feed Water
															Front	Back	Side		
		Ft. In.	Ft.In.	Ft.In.	Ft. In.	Ft.In.	Ft. In.	Ft. In.	Ft.In.	Ft. In.	Ft. In.	In.	In.	In.	In.	In.	In.	In.	In.
10	1577	21-9 1/8	8 -0	17 -0	5 -0	6	12 -9 1/2	1- 4 29/32	2 - 4	5 -2 1/2	4- 5 19/32	42	42	48	18	13	18	5	2
13	2051	21-9 1/8	9 -6	17 -0	6 -6	6	12 -9 1/2	1- 4 29/32	2 - 4	5 -2 1/2	4- 5 19/32	42	42	48	18	13	18	5	2
16	2524	21-9 1/8	11 -0	17 -0	8 -0	6	12 -9 1/2	1- 4 29/32	2 - 5	5 -2 1/2	4- 5 19/32	42	42	48	18	13	18	6	2 1/2
19	2997	21-9 1/8	12 -6	17 -0	9 -6	6	12 -9 1/2	1- 4 29/32	2 - 5	5 -2 1/2	4- 5 19/32	42	42	48	18	13	18	6	2 1/2
22	3470	21-9 1/8	14 -0	17 -0	11 -0	6	12 -9 1/2	1- 4 29/32	2 - 5	5 -2 1/2	4- 5 19/32	42	42	48	18	13	18	6	2 1/2
27	4259	21-9 1/8	17 -6	17 -0	13 -6	6	12 -9 1/2	1- 4 29/32	2 - 5	5 -2 1/2	4- 5 19/32	42	42	48	18	13	18	6	3
14	2401	24-1 5/8	10 -0	17 -9 1/2	7 -0	7	14 -2	1- 2 3/32	2 - 5	5 -2 1/2	4- 8 3/8	42	42	48	18	13	18	6	2
20	3544	24-1 5/8	13 -0	17 -9 1/2	10 -0	7	14 -2	1- 2 3/32	2 - 5	5 -2 1/2	4- 8 3/8	42	42	48	18	13	18	6	2 1/2
26	4607	24-1 5/8	16 -0	17 -9 1/2	13 -0	7	14 -2	1- 2 3/32	2 - 5	5 -2 1/2	4- 8 3/8	42	42	48	18	13	18	6	3
29	5139	24-1 5/8	17 -6	17 -9 1/2	14 -6	7	14 -2	1- 2 3/32	2 - 5	5 -2 1/2	4- 8 3/8	42	42	48	18	13	18	6	3 1/2
12	2896	25-7 5/8	9 -0	18 -4 7/16	6 -0	7 -6	15 -2	1- 0 1/8	2 - 5	5 -2 1/2	4-10 3/8	42	42	48	18	13	18	6	2
18	3444	25-7 5/8	12 -0	18 - 4 7/16	9 -0	7 -6	15 -2	1- 0 1/8	2 - 5	5 -2 1/2	4-10 3/8	42	42	48	18	13	18	6	2 1/2
24	4592	25-7 5/8	15 -0	18 -4 7/16	12 -0	7 -6	15 -2	1- 0 1/8	2 - 5	5 -2 1/2	4-10 3/8	42	42	48	18	13	18	6	3
10	1967	26-6 1/8	8 -0	18 -7 1/16	5 -0	8	15 -6 1/2	0-11 3/8	2 - 4	5 -2 1/2	4-11 1/8	42	42	48	18	13	18	5	2
22	4336	26-6 1/8	14 -0	18 -7 1/16	11 -0	8	15 -6 1/2	0-11 3/8	2 - 5	5 -2 1/2	4-11 1/8	42	42	48	18	13	18	6	3
28	5509	26-6 1/8	17 -0	18 -7 1/16	14 -0	8	15 -6 1/2	0-11 3/8	2 - 5	5 -2 1/2	4-11 1/8	42	42	48	18	13	18	6	3 1/2
30	5902	26-6 1/8	18 -0	18 -7 1/16	15 -0	8	15 -6 1/2	0-11 3/8	2 - 5	5 -2 1/2	4-11 1/8	42	42	48	18	13	18	6	3 1/2
12	2288	23-9 1/8	9 -6	18 -3 3/8	6 -0	7	13 -6	1-10 11/16	2 - 8	5 -4 1/2	4- 8 11/16	48	48	54	22	18	22	6	2
15	2765	23-9 1/8	11 -2	18 -3 3/8	7 -6	7	13 -6	1-10 11/16	2 - 8	5 -4 1/2	4- 8 11/16	48	48	54	22	18	22	6	2 1/2
27	5013	23-9 1/8	17 -2	18 -3 3/8	13 -6	7	13 -6	1-10 11/16	2 - 8	5 -4 1/2	4- 8 11/16	48	48	54	22	18	22	6	3 1/2
35	6499	23-9 1/8	21 -2	18 -3 3/8	17 -6	7	13 -6	1-10 11/16	2 - 8	5 -4 1/2	4- 8 11/16	48	48	54	22	18	22	8	3 1/2
13	2933	27-3 1/8	10 -2	19 -8 11/16	6 -6	8	16 -2	1- 7	2 - 8	5 -4 1/2	5- 0 3/8	48	48	54	22	18	22	6	2 1/2
16	3610	27-3 1/8	11 -6	19 -8 11/16	8 -0	8	16 -2	1- 7	2 - 8	5 -4 1/2	5- 0 3/8	48	48	54	22	18	22	6	3
20	4768	27-3 1/8	14 -6	19 -8 11/16	11 -0	8	16 -2	1- 7	2 - 8	5 -4 1/2	5- 0 3/8	48	48	54	22	18	22	6	3 1/2
24	5798	27-3 1/8	17 -2	19 -8 11/16	13 -6	8	16 -2	1- 7	2 - 8	5 -4 1/2	5- 0 3/8	48	48	54	22	18	22	6	3
15	3982	31-9 1/8	11 -2	21 -2	7 -6	10	18 -6	1- 3 5/16	2 - 8	5 -4 1/2	5- 4 1/16	48	48	54	22	18	22	6	3
30	7963	31-9 1/8	18 -6	21 -2	15 -0	10	18 -6	1- 3 5/16	2 - 8	5 -4 1/2	5- 4 1/16	48	48	54	22	18	22	8	4
36	10087	31-9 1/8	22 -6	21 -2	19 -0	10	18 -6	1- 3 5/16	2 - 8	5 -4 1/2	5- 4 1/16	48	48	54	22	18	22	8	2-3 1/2
42	11149	31-9 1/8	24 -6	22 -2	21 -0	10	18 -6	1- 3 5/16	2 - 8	5 -4 1/2	5- 4 1/16	48	48	54	22	18	22	8	2-3 1/2

TABLE V. LONGITUDINAL STRAIGHT TUBE BOILER AND SETTING DATA OF THE GENERAL TYPE AS SHOWN BY FIG. 5

Diameter of Drum	Size Tube Bank		Number of Tubes	Square Feet Heating Surface for the Various Tube Lengths Ft.					Height Uptake - Boiler set 72 in. Floor to Front Header	Over all widths 18 in. side and 22 in. division walls of brick		Width of grates	Brick for 18 ft. tube boiler - hand fired, 18 in. side and 22 in. division walls, 4½ in. F. B. Lining				Approximate weight of 18 ft. tube boilers, 160 lb. pressure	
	Vertical	Horizontal								Single Set	Battery Set		Single setting		Battery setting		Bare (Heavy Piece)	Total (With Fixtures and Fittings
In.	Ft.	Ft.		16	17	18	19	20	Ft. In.	Ft. In.	Ft. In.	Ft.In.	Fire	Red	Fire	Red		
30	2½	4	43	795	846	894	942	1000	13 - 1	7 - 6	13 - 10	4 - 6	2080	16,120	4160	25,400	10,589	17,500
30	2½	4½	51	956	1016	1073	1131	1184	13 - 7	7 - 6	13 - 10	4 - 6	2180	16,540	4860	26,000	12,300	19,500
36	2½	5	57	1053	1111	1180	1244	1304	14 - 1	8 - 0	14 - 10	5 - 0	2330	16,600	4660	26,100	14,422	22,000
36	2½	5	63	1167	1236	1309	1380	1446	14 - 1	8 - 6	15 - 10	5 - 6	2465	17,000	4930	26,800	15,484	24,500
36	2½	5½	69	1270	1348	1425	1502	1574	14 - 1	9 - 0	16 - 10	6 - 0	2500	17,300	5000	27,500	16,597	26,500
36	3	5	74	1354	1436	1520	1602	1688	14 - 7	8 - 6	15 - 10	5 - 6	2580	17,600	5300	29,000	17,394	27,000
36	3	5	74	1354	1436	1520	1602	1688	14 - 7	8 - 6	15 - 10	5 - 6	2650	18,400	5300	29,000	17,394	27,000
36	3	5½	81	1472	1564	1650	1740	1821	14 - 7	9 - 0	16 - 10	6 - 0	2725	18,800	5450	29,900	18,602	29,500
42	3	6	88	1602	1709	1804	1901	1990	15 - 1	9 - 6	17 - 10	6 - 6	2855	19,600	5710	31,100	20,023	32,000
42	3	6½	95	1739	1844	1949	2054	2157	15 - 1	10 - 0	18 - 10	7 - 0	2985	20,400	5970	32,600	22,698	35,500
42	3	6½	98	1792	1888	2005	2136	2214	15 - 1	10 - 3	19 - 4	7 - 3	3106	20,900	6212	33,800	22,940	36,519
42	3	7½	109	1973	2094	2213	2323	2453	15 - 1	11 - 0	20 - 10	8 - 0	3180	21,900	6360	35,600	25,016	39,500
48	3	8	116	2106	2237	2362	2484	2606	15 - 7	11 - 6	21 - 10	8 - 6	3230	22,600	6460	36,700	27,290	43,500
48	3	7½	124	2250	2382	2522	2658	2794	16 - 1	11 - 0	20 - 10	8 - 0	3275	22,600	6550	35,900	28,910	44,000
48	3½	8	132	2389	2534	2678	2822	2966	16 - 1	11 - 6	21 - 10	8 - 6	3350	23,100	6700	36,800	30,667	47,000
48	3½	8½	140	2528	2681	2834	2983	3139	16 - 1	12 - 0	22 - 10	9 - 0	3425	23,600	6850	37,100	31,698	49,000
48	4	8	149	2678	2852	3004	3164	3324	16 - 7	11 - 6	21 - 10	8 - 6	3470	23,600	6940	37,700	33,281	50,000
36	4	8½	158	2876	3052	3226	3401	3576	15 - 7	12 - 0	22 - 10	9 - 0	3495	24,000	6990	39,200	35,105	53,000
36	4	9	167	3009	3195	3379	3562	3744	15 - 7	12 - 6	23 - 10	9 - 6	3530	24,100	7060	39,600	36,925	56,000
36	4½	8½	175	3173	3356	3560	3752	3946	16 - 1	12 - 0	22 - 10	9 - 0	3595	24,200	7190	39,500	38,000	60,000
36	4½	9	185	3345	3550	3752	3957	4159	16 - 1	12 - 6	23 - 10	9 - 6	3625	24,700	7250	40,400	40,000	62,000
36	5	8½	198	3571	3789	4006	4224	4440	16 - 7	12 - 3	23 - 4	9 - 3	3657	25,200	7314	41,200	42,582	64,782
36	4½	10½	215	3859	4095	4330	4564	4799	16 - 1	14 - 0	26 - 10	11 - 0	4900	23,400	11,800	42,000	50,662	72,061
42	4½	11	225	4065	4317	4564	4811	5059	16 - 7	14 - 6	27 - 10	11 - 6	4916	23,500	11,832	42,200	54,626	76,512
42	5	11	248	4433	4737	5001	5262	5518	17 - 1	14 - 6	27 - 10	11 - 6	6000	24,000	12,000	41,000	58,000	80,000
42	4½	12½	250	4510	4784	5058	5331	5604	16 - 7	15 - 9	30 - 4	12 - 9	5800	26,000	11,600	43,000	58,000	80,000
42	5	12½	275	4928	5228	5527	5828	6127	17 - 1	15 - 9	30 - 4	12 - 9	5884	26,600	11,768	42,156	61,382	83,488
42	5	13½	303	5407	5736	6065	6395	6723	17 - 1	17 - 0	32 - 10	14 - 0	5984	28,720	11,968	43,124	66,180	86,976
42	5	14½	330	5877	6234	6591	6949	7305	17 - 1	18 - 3	35 - 4	15 - 3	6084	26,640	18,168	44,292	69,562	90,464
42	5	15½	352	6240	6637	7027	7398	7773	17 - 1	19 - 3	37 - 4	16 - 3	6147	28,940	12,294	46,292	72,562	93,464
42	5½	15½	384	6772	7219	7633	8048	8460	17 - 7	19 - 3	37 - 4	16 - 3	6231	29,540	12,462	49,342	77,365	98,397
42	6	15½	410	7216	7686	8127	8569	9008	18 - 1	19 - 0	36 - 10	16 - 0	6315	30,140	12,630	50,392	82,168	103,330

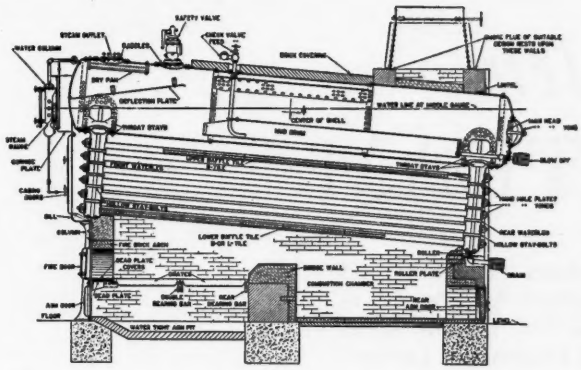


FIG. 5. DETAILS OF A LONGITUDINAL DRUM STRAIGHT TUBE BOILER

superheaters. It is impossible to lay down definite limits of each type. This has been done approximately in Fig. 2, the shaded portion of the chart showing pressures and capacities available in standard equipment or equipment that has already been built. The largest boiler built to date can develop over 1,250,000 lb. per hr. of steam[5] while the highest pressure commercial installation is the 3200 lb. Benson boiler.[6]

The simplest of the so-called power boilers is the well known and widely used horizontal return tubular boiler shown in cross section of Fig. 3. The allowable pressure depends upon the thickness and quality of the metal in the shell, the type of joint and upon the factor of safety. In general

$$P = (T \times t \times E) \div (R \times F)$$

where

P = Allowable working pressure, lb. per sq. in.
T = Tensile strength lb. per sq. in. (55,000 to 65,000)
t = Plate thickness in in.
E = Efficiency of joint (or ligaments between tube holes if lower)
R = Radius of shell in in.
F = Factor of safety (= 5 for new work)

A detailed discussion of this phase of boiler work is beyond the scope of this article. It has been well standardized, however, and is completely covered by the Boiler Code.[7]

Ratings of this type of boiler are usually figured on the basis of 10 sq. ft. of heating surface per boiler horsepower, the heating surface being considered the inside tube area in contrast to water tube boilers when the outside tube area is used. The complete heating surface is obtained by calculation and is equal to the inside area of the tubes plus one-half the cylindrical part of the shell plus ⅔ of the rear head minus the combined cross-sectional area of the tubes. Data for typical boilers are given by Tables I and II. A typical setting drawing with various parts of the boiler identified is shown by Fig. 3.

Limitations of this type boiler have led to a number of adaptions of modern designs to be used in the same

[5]Power Plant Engineering, p. 17, January 1, 1932.
[6]Progress with the Benson Boiler, Power Plant Engineering, p. 378, September 1933.
[7]Boiler Construction Code published by the American Society of Mechanical Engineers, 29 W. 39th St., New York, N. Y., particularly Sec. I, Power Boilers, Sec. II, Material Specifications; Sec. VI, Rules for Inspection, and Sec. VII, Suggested Rules for Care of Power Boilers.

TABLE VI. DATA FOR STRAIGHT TUBE CROSS DRUM BOILERS AS SHOWN BY FIG. 4C

M Drum diam. In.	Tubes High	Tubes Wide	Heating Surface	A Ft.	A In.	Width of Setting Ft.In.	Width of Furnace K Ft.In.	F Ft.	F In.	H Ft.	H In.	Steam Outlet In.	Feed Water In.
	7	11	1572	20-	1 3/4	8 -6	5 -6	9-	7 1/4	2 -	1 5/16	5	2
	7	12	1720	20-	1 3/4	9 -0	6 -0	9-	7 1/4	2 -	1 5/16	5	2
	7	14	2015	20-	1 3/4	10 -0	7 -0	9-	7 1/4	2 -	1 5/16	5	2
42	9	16	2834	20-	4 7/8	11 -0	8 -0	10-	6 3/4	2 -	4 7/16	6	2 1/2
	9	20	3560	20-	4 7/8	13 -0	10 -0	10-	6 3/4	2 -	4 7/16	6	2 1/2
	15	13	3558	21-	2 1/4	9 -6	6 -6	13-	5 1/2	3 -	1 13/16	6	2 1/2
	9	21	3743	20-	4 7/8	13 -6	10 -6	10-	6 3/4	2 -	4 7/16	6	3
	9	22	3925	20-	4 7/8	14 -0	11 -0	10-	6 3/4	2 -	4 7/16	6	3
	10	21	4080	20-	5 7/16	13 -6	10 -6	11-	0 9/16	2 -	6	6	3
	10	24	4675	20-	5 7/16	15 -0	12 -0	11-	0 9/16	2 -	6	6	3
48	10	25	4874	20-	5 7/16	15 -6	12 -6	11-	0 9/16	2 -	6	6	3 1/2
	10	26	5072	20-	5 7/16	16 -0	13 -0	11-	0 9/16	2 -	6	6	3 1/2
	15	20	5536	21-	2 1/4	13 -0	10 -0	13-	5 1/2	3 -	1 13/16	6	3 1/2
	15	21	5819	21-	2 1/4	13 -6	10 -6	13-	5 1/2	3 -	1 13/16	6	3 1/2
	12	26	5988	20-	9 9/16	16 -0	13 -0	12-	0 3/16	2 -	9 1/8	8	3 1/2
	12	28	6392	20-	9 9/16	17 -0	14 -0	12-	0 3/16	2 -	9 1/8	8	3 1/2
	13	27	6616	20-11	1/8	16 -6	13 -6	12-	5 15/16	2 -10	11/16	8	3 1/2
	13	28	6865	20-11	1/8	17 -0	14 -0	12-	5 15/16	2 -10	11/16	8	3 1/2
	13	29	7114	20-11	1/8	17 -6	14 -6	12-	5 15/16	2 -10	11/16	8	4
54	15	26	7227	21-	2 1/4	16 -0	13 -0	13-	5 1/2	3 -	1 13/16	8	4
	15	27	7510	21-	2 1/4	16 -6	13 -6	13-	5 1/2	3 -	1 13/16	8	4
	15	28	7793	21-	2 1/4	17 -0	14 -0	13-	5 1/2	3 -	1 13/16	8	4
	16	27	7948	21-	3 13/16	16 -6	13 -6	13-11	5/16	3 -	3 3/8	8	4
	16	28	8248	21-	3 13/16	17 -0	14 -0	13-11	5/16	3 -	3 3/8	8	4
	16	29	8547	21-	3 13/16	17 -6	14 -6	13-11	5/16	3 -	3 3/8	8	5

head room. Figure 1A is a Dutch oven setting of an inexpensive type with partially water cooled furnace, and two-pass boiler; Fig. 1B provides a self-contained water cooled furnace; Fig. 1C is a water cooled furnace type in which the gases of combustion flow longitudinally with the drums in contrast to Fig. 1D, also a two drum boiler, which might be called a cross drum type; Fig. 1E is a three drum waste heat boiler, the gases flowing longitudinally with the drums.

In addition to these more or less special designs there is the popular low head room type shown by Fig. 4A, especially designed to fit into the same head room and replace h.r.t. units. Data on this type are given in Table III while data on a three drum bent tube boiler of standard type as shown by Fig. 6B are given by Table IV. Straight tube boilers are built in the longitudinal and cross drum type, Fig. 5 showing details of first and Fig. 4C the arrangements of the second. Data on this latter type are given by Table VI.

Efficiency calculations are ordinarily made from steam, water and coal measurements although it may be approximated by calculating the losses[8] and subtracting them from 100 per cent. In general the overall boiler and furnace efficiency can be calculated by the following formula:

$$E = 100\,W\,(H_2 - H_1) \div CB$$

where

E = Boiler and furnace efficiency in per cent
W = Steam or water flow in lb. per hr.
H_1 = Total heat in water entering boiler or economizer, B.t.u. per pound
H_2 = Total heat of steam leaving boiler or superheater, B.t.u. per pound
C = Coal in lb. per hr.
B = Heating value of coal in lb. per hr.

Methods and calculations of tests with suitable log sheets are given in detail in the A.S.M.E. Power Test Code on Stationary Steam Generating Units.

Boiler tests should be made with extreme accuracy as even under the best conditions the limits of accuracy may reasonably be taken as plus or minus 3 per cent.

[8]Boiler Test by the Heat Balance Method, by B. J. Cross, Sept. 15, 1929, p. 1004.

Solving the Problem of WET STEAM

Moisture in Steam Reduces Superheat and Carries Over Impurities which Are Abrasive to Metal Surfaces and Often Coat Turbine Blades So that the Capacity and Efficiency Is Lowered

MOISTURE IN STEAM, which in practice ranges from a very fine mist of ¼ to ½ per cent of the total steam weight to foaming and priming where practically all the water in the boiler drum may be carried over to the steam line, depends upon a number of factors all of which are susceptible to control, although not all are as yet thoroughly understood. Some of these factors are: boiler design; rate of evaporation; condition of boiler water and position of water level.

The importance of clean dry steam is apparent. Moisture carried over into the superheater reduces superheat as shown by Fig. 1, this reduction for a given quality steam being much more for low than for high pressure installations. Its importance in the lower pressure ranges is greater because the superheat is usually lower and can be ill afforded. The effect is much more pronounced with engines than with turbines, one test reported on a triple expansion engine showed a reduction in water rate of 5.88 per cent when steam quality was improved from 97.4 to 99.83 per cent. At 150 lb. this increase in quality would have meant an increase of about 25 deg. F. in superheat. The decrease in water rate would have been about 2.33

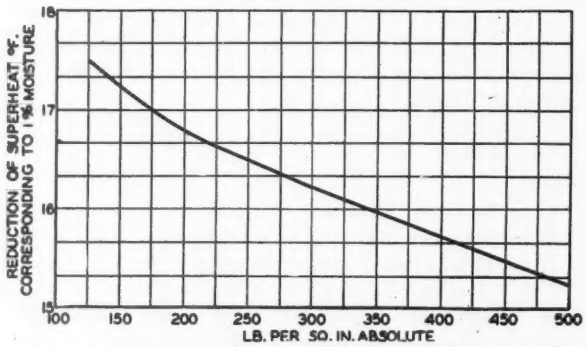


FIG. 1. EACH PER CENT MOISTURE IN THE STEAM REDUCES THE SUPERHEAT FROM 15 TO 20 DEG. F. BY MAKING A BOILER OUT OF THE SUPERHEATER

per cent for every 10 deg. F. in superheat. A turbine test showed a reduction of about 0.875 per cent in water rate for every 10 deg. F. increase in superheat.

Besides reducing superheater capacities and prime mover efficiency, the moisture carries over suspended and dissolved solids which build up deposits in pipes and on turbine blades[1] or acts as an abrasive to increase wear on pipes, valves, cylinders and blades. In excessive quantities this moisture causes dangerous water hammer. Boiler water treatment,[2] priming and foaming[3] and water level problems[4] are beyond the scope of this article which deals primarily with the question of dry steam production as a boiler problem.

[1]Operating Experience With 1300-lb. Pressure, by John Anderson, Power Plant Engineering, Feb. 15, 1928, p. 229.

Drum Baffling

The conventional boiler must provide for the entrance of feedwater, the removal of steam and blowdown. The first requirement is often fulfilled in the longitudinal drum boiler by admitting the cold feedwater through a pipe which enters the drum and discharges well into the drum past the front header and

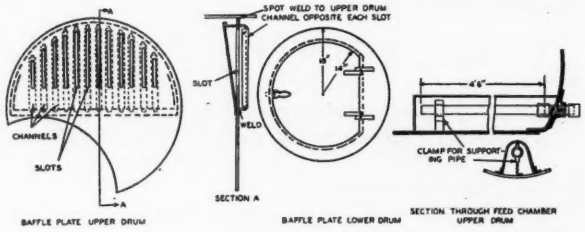


FIG. 2. DRUM BAFFLING IN A TWO DRUM BENT TUBE BOILER

in the direction of the natural circulation. In cross drum and bent tube boilers the feedwater conventionally enters one or both ends and discharges into a trough which overflows as a weir to give even distribution. This idea extended further in some bent tube boilers where feedwater enters the compartment in the rear drum, flows downward through the rear tubes and is reversed by the compartment in the lower drum returning through the middle section tubes to the top drum where it enters the ring circulation of the boiler. The effect of the rear group of tubes is like an economizer and the second drum is fitted with a blowdown connection so as to eliminate solids thrown down by heat absorbed in the first downward pass. Feed pipe baffles in cross drum boilers are shown at the bottom of the drums in Fig. 3.

Baffles Must Be Tight

Baffling the drum for separating moisture from the steam is of course much more difficult than baffling

[2]For a comprehensive treatise of boiler drum baffling in relation to boiler water see Studies of Moisture at High Rates of Evaporation by A. R. Mumford. A.S.M.E. Fuels & Steam Power, Vol. 51, No. 22, 1929.

[3]Power Plant Engineering, May 1, 1931, p. 528.

[4]Boiler Water Level Problems, N.E.L.A. Publication 241, January, 1933.

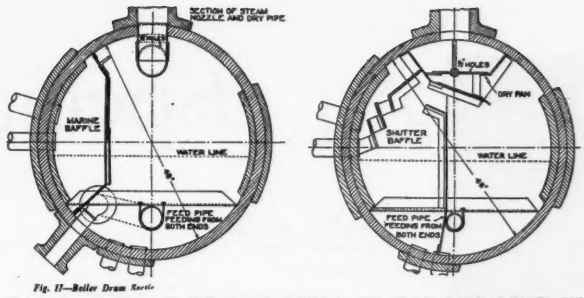


FIG. 3. DETAILS OF TWO TYPES OF BAFFLES USED IN CROSS DRUM BOILERS. THE ONE AT THE RIGHT IS FOR HIGH CAPACITY BOILERS

for controlling the direction of flow of the water. Not only must the baffle be correctly designed but it must be tight. Figure 2 shows an unusually simple and effective steam baffle used in a two drum boiler.[5] One end of the drum is baffled off as an economizer section and it is from this comparatively quiet end that the steam is taken after passing through the grill in the steam separator baffle. The segment cut out at the bottom is to allow free circulation of the water.

A type of baffle widely used with cross drum boilers and sometimes called the marine type is built as shown at the left of Fig. 3. This consists of a vertical baffle placed somewhat back of, and away from, the steam discharge tubes coming in at the left. Entering steam and water are separated beneath the baffle, travel lengthwise of the drum under the baffle. At the end the steam reverses and again travels lengthwise of the drum to the steam nozzle.

High Capacity Requires Changes

Above about 150,000 lb. per hr. capacity the velocity of steam behind the baffle is said to be sufficiently[6] high to prevent good separation or even to actually pick up water and carry it along. To remedy this condition a fundamental change was made in the larger units by substituting transverse for longitudinal steam flow with a shutter type of baffle shown at the left side of the drum at the right of Fig. 3. In front of the steam circulators is a set of inclined or shutter baffles against which the steam and water impinge. The water deflected downward is separated from the steam which is directed upward by a typical vertical baffle with an inclined top portion located outside the shutter baffle. Above this is another inclined baffle which forms a channel for the steam flow and practically reverses its direction. After passing the second inclined baffle the flow is again reversed and led into a box type dry pipe which replaces the conventional round dry pipe.

Comparative tests reported[7] by an operating company showed that in general steam which had passed through the shutter type baffle carried lower solid content than that leaving the marine type baffle. With 5000 p.p.m. concentration steam leaving the shutter type baffle carried no more solids than steam from the marine baffle at 2500 p.p.m. Only with high concentration and high varying water levels did an increased carry over occur with the shutter baffle.

[5]New Type Boiler at Caron Spinning Co., Power Plant Engineering, Feb. 15, 1932, p. 160.
[6]N.E.L.A. Proceedings, 1929, p. 1300.
[7]N.E.L.A. Proceedings, 1930, p. 887.

An interesting type of scrubber[8] was recently installed at South Amboy Station to replace baffling which was allowing a carryover of only 8 to 10 p.p.m. This is ordinarily considered good operation so that the efficiency of the new scrubber, which brought the solid content of the steam down to 0.2 p.p.m. can hardly be questioned.

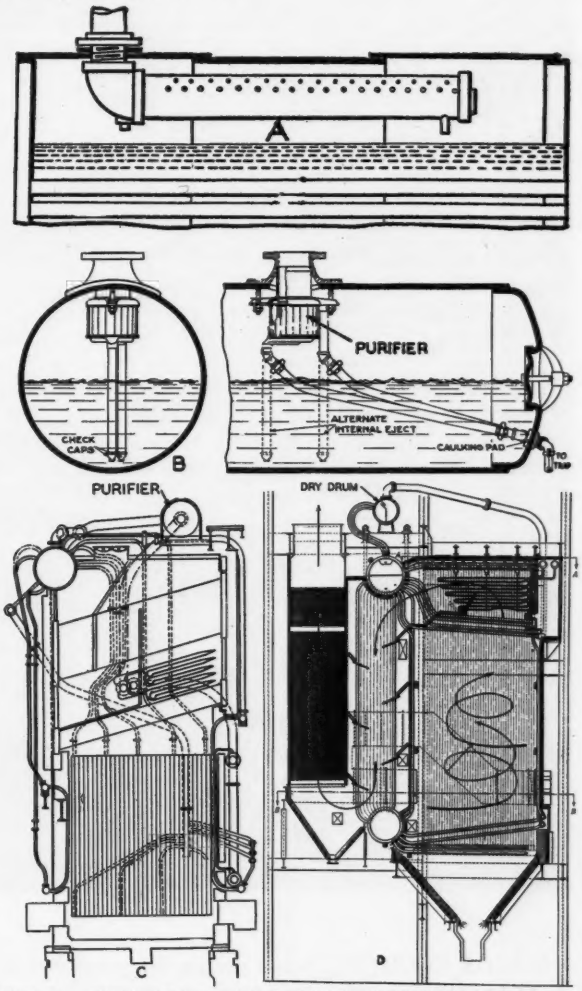


FIG. 4. METHODS OF IMPROVING STEAM QUALITY. A, THE CONVENTIONAL DRY PIPE. B, THE INTERNAL SEPARATOR. C, THE EXTERNAL SEPARATOR. D, ONE TYPE OF DRY DRUM

These arrangements are of course all variations or additions to the conventional round pipe perforated dry pipe as shown by Fig. 4A. In existing boilers of the usual size and design, the same and sometimes better, performance can be obtained by installing separators either in the boiler drums as shown by Fig. 4B or externally as shown by Fig. 4C. Water separated from the steam can be returned either to the boiler drum or to the blowoff tank direct or through a heat exchanger.

Another variation of the external separator is the so-called dry drum, one arrangement of which is shown by Fig. 4D. Normally this drum is dry and serves only as a separating chamber, the water being returned to the boiler.

[8]Design and Operating Experience at South Amboy Station, by R. C. Roe. A.S.M.E. Metropolitan Section Meeting, April 20, 1933.

Special

A Large European Loeffler Boiler Installation

IN THIS COUNTRY, boiler design for pressures up to 1400 lb. has been along more or less conventional straight tube and bent tube construction as first used by the pioneer high pressure stations, Edgar and Lakeside. Two notable exceptions are the Philip Carey boilers[1] and the Yarnall-Waring boiler[2]. The first is an unusual bent tube design for 1800 lb. with the headers, one horizontal and the other vertical, 90 deg. apart with the tubes connecting them approximating a quarter circle. The second unit for 2500 lb. for test or laboratory service is also for natural circulation. Both the drum and header are outside of the setting, the heat absorbing surface forming a sinuous path up through the furnace, the arrangement being something like the usual convection superheater.

Interest in boilers for pressures in excess of 1500 lb. appears to be in the direction of forced circulation, however, some notable advances having been made in Europe along these lines. Perhaps those of greatest commercial interest being the Benson critical pressure boiler and the Loeffler steam circulation boiler. The Benson boiler consists fundamentally[3] of a single tube into which water is pumped at critical pressure and progressively heated until it vaporizes without boiling. The development of this boiler into a commercial unit with superheater, reheater and economizers is shown by Fig. 1A. An important recent development in this unit being its operation at variable pressure below the critical, thus doing away with governing at the turbine, the boiler pressure being a function of the load.

In this country the series boiler has been developed somewhat differently, Fig. 1B showing an experimental design with seven tubes in series. Water enters at A, flows downward through the economizer section in the stack, comes out at B near the middle, enters the bottom at C, flows upward when it is vaporized and superheated, steam coming out at D. The superheat is determined by the proportioning water heating and steam heating surface in the boiler section, in effect by controlling the water level. This is the type of boiler installed at Purdue[4] for experimental use. Another series boiler which deserves mention, not because of high pressure but for its special use, is the Dow Chemical Co.[5] unit using a diphenyl preparation instead of water. This unit, shown by Fig. 1C, has the heating surface totally enclosing the furnace with the outlet section at the top.

Steam Circulating Pump Used

In Europe the Loeffler boiler[6] is in wide use for pressures up to 1900 lb. The operation of this boiler is quite unconventional, the water being boiled or vaporized in a drum outside and separate from the boiler, as shown by Fig. 2. Water is pumped by the feed pump E through the economizer F to the vaporizing drum A. Steam from A is pumped by the steam

[1]Power Plant Engineering, p. 15, Jan. 1, 1933.
[2]2500 lb. testing boiler, Power Plant Engineering, p. 1153, Dec. 1, 1931.
[3]Recent developments of the Benson boiler by H. Gleichmann Power Plant Engineering, p. 612, June 1, 1930; also p. 378 September, 1933.
[4]A.S.M.E. Transactions RP-54-1b and 1a, 1931.
[5]Dowtherm Vapor for Process Heating, Power Plant Engineering, p. 787, November, 1932.
[6]All This Boiler is Superheater, by J. Havlicek, Power Plant Engineering, p. 588, May 15, 1929.

and Super-pressure Boilers *for* Industry

pump or compressor B through the superheater C out through line D. Some of this superheated steam is used for the prime mover and part is circulated back through drum A where the superheat is utilized to vaporize the water and make more saturated steam for pump B. A recent unit of this type designed for 175,000 lb. per hr. of 1900 lb., 950 deg. F. steam and installed at Trebovice Station, Czecho-Slovakia, is shown by the headpiece of this article. Two units for a Moscow Station are designed for twice this capacity.

Another design using superheated steam to generate additional saturated steam outside of the boiler is made by Brown-Boveri[7]. In this, steam from an auxiliary starting boiler is superheated and used to vaporize more steam in an external drum or vessel. This steam is then superheated and the process repeated, the volume of steam increasing each step. No steam pump is used, a sufficient pressure drop being used between stages to cause the necessary steam flow. A more recent development of this company is the Velox[8] pressure boiler shown by Fig. 3.

Two systems of these steam generators are designed; steady pressure and explosion types. For the explosion system, the combustion chamber is periodically charged by the compressor with an explosive fuel-air mixture, the inlet valves closed and the mixture ignited. Pressure rises to 4 to 5.5 times charging pressure. After the explosion (really after total combustion), the gases of combustion flow through an outlet valve into the heating tubes, evaporating the water which flows at high velocity outside of these tubes. After passing through the tubes at high velocity, the gases are checked before the nozzles of the gas turbine so that they have a large pressure drop to atmosphere. In order to obtain uniform output of the gas turbine, several combustion chambers are arranged.

For the steady-pressure system, the combustion chamber is also loaded with a compressed fuel-air mixture which, however, is not explosive. Pressure in the chamber is kept steady by a compressor driven by a gas turbine, which has to be arranged intermediate in the gas flow so that heating surfaces are both before and after the gas turbine. The illustration Fig. 3 shows the general layout of a steady-pressure generator with gas-turbine-driven compressor.

Still different is the revolving boiler shown by Fig. 4, feedwater entering the rotor is forced by centrifugal action out in the arm a. The difference in centrifugal force of the water in arm a over that of the stream in the arm b determines the pressure which is of course a function of the speed. A 49.5 in. dia. unit operating at a speed of 3000 r.p.m. will develop a pressure of about 1700 lb.

The revolving principle has been used successfully in the Atmos[9] boiler for several years. The rotating

[7]N.E.L.A. Proceedings 1929, p. 1244.
[8]Pressure Fired Boilers, by W. G. Noack, Power Plant Engineering, p. 119, March 1933.

[9]N.E.L.A. Proceedings 1931, p. 809-17.

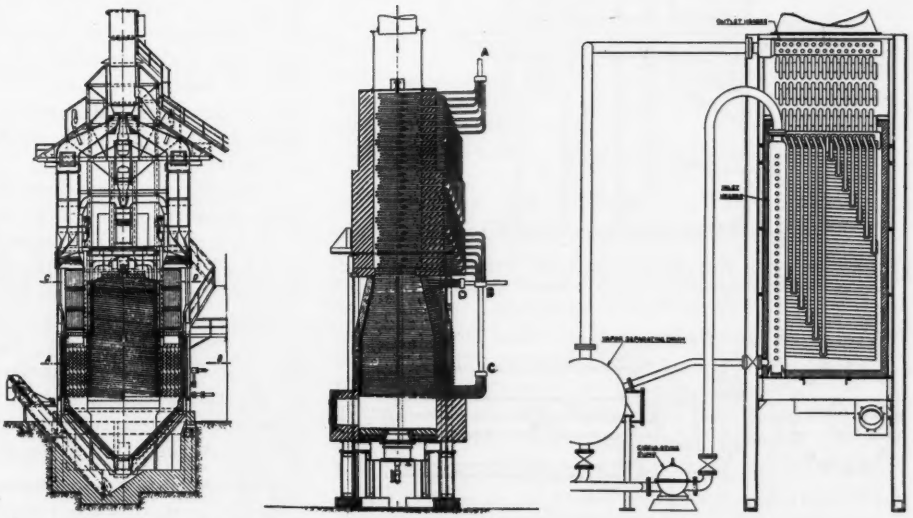


FIG. 1. THREE TYPES OF FORCED CIRCULATION BOILERS

A. The latest Benson Boiler for critical pressure operation. B. A series boiler of the type used at Purdue good for 500 lb. C. A special boiler using a diphenyl compound instead of water

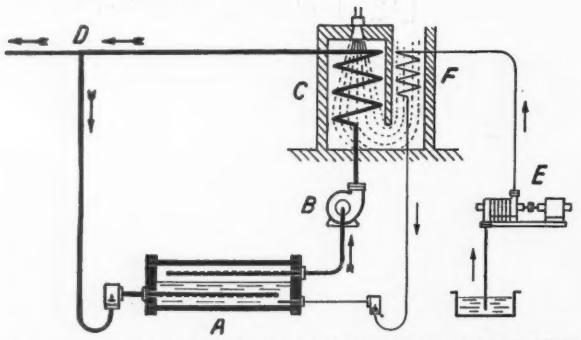


FIG. 2. DIAGRAMMATIC ARRANGEMENT OF THE LOEFFLER BOILER IN WHICH STEAM IS PUMPED THROUGH THE HEATING COILS IN THE FURNACE

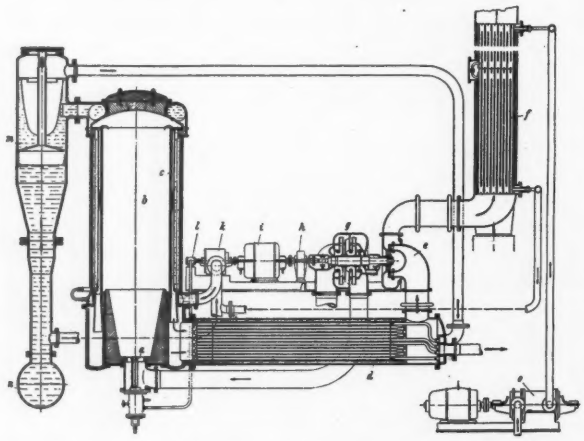


FIG. 3. THE VELOX BOILER IN WHICH THE FURNACE IS UNDER A PRESSURE OF 80 TO 180 LB. ABS.

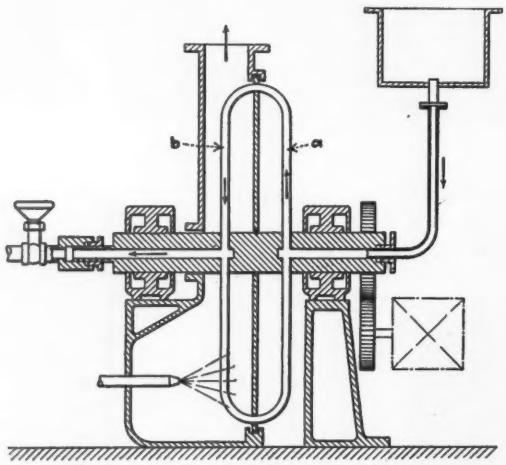


FIG. 4. IN THIS REVOLVING BOILER PRESSURE IS DEVELOPED BY CENTRIFUGAL FORCE

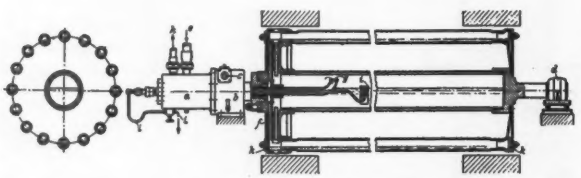


FIG. 5. THE ATMOS BOILER IN WHICH THE TUBES ALSO REVOLVE

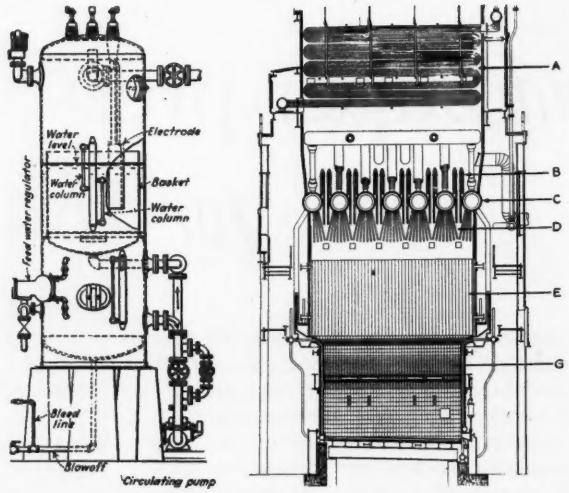


FIG. 6. A. ELECTRIC BOILERS ARE FINDING CONSIDERABLE USE. B. MERCURY BOILER AS INSTALLED AT SCHENECTADY

tubes of a recent unit are shown by Fig. 5. This tube bundle is set over a furnace in much the same manner as an an h.r.t. boiler. Water enters the rotors after passing through a steaming economizer. Water to the rotors is controlled automatically and is distributed evenly around the inner periphery by centrifugal force. The center of the tube is free for the liberation of steam which is delivered out of the pinion to the superheater. A test reported showed an evaporation of over 60 lb. per sq. ft. of boiler surface.

Many adaptions of more or less conventional design have been made in Europe and of these the 1700-lb Ilse[10] boilers are perhaps representative. These are two drum bent tube boilers with both drums in the same vertical plane. Another type which differs some from the usual practice is the Schmidt-Hartmann[9] unit as installed at Bitterfeld, Germany. The drum and header is located outside of the boiler setting with the heating surface arranged as in the Yarnall-Waring boiler. Steam generated in this section is used to generate more steam in an outside vessel as in the Loeffler design, the generating steam condensing and flowing back to the boiler. The boiler circuit being completely closed and filled with pure water, difficulties with feedwater treatment and scale are eliminated. This closed system has recently been used in a process boiler in this country[11]. The boiler operated directly under the process coils needs no feed pump, the condensed steam flowing back to the boiler by gravity as in a heating boiler.

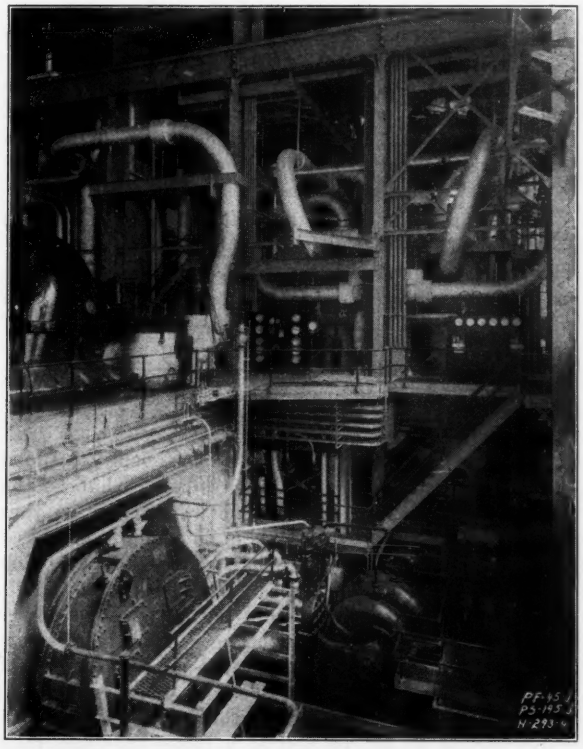
Two other boilers deserve mention, not because of high pressure but because of their special design, the electric boiler and the mercury boiler[12] both shown by Fig. 6. The mercury boiler is for three phase current. the load being controlled by varying the depth of immersion of the electrodes by pumping water into or out of the upper basket. The mercury boiler[11] which has just entered its commercial phase was developed at Hartford and the two stations just completed, Schenectady and Kearny, are intended to bring out the full possibilities of the binary cycle.

[10] 1700 lb. plant of novel design, Power Plant Engineering, p. 1058, November 1, 1931.

[11] 20 yr. of Mercury Vapor Development, p. 263, June 1933 and Features of Outdoor Mercury Vapor Plant, p. 354, August 1933 Power Plant Engineering.

Superheat *is Responsible for* Higher Efficiencies

High Temperatures Made Possible By Superheaters and Reheaters Play Important Roles in Efficiency Gains in the Modern Power Station Using a Simple or Reheat Cycle



STEAM REHEATERS IN A MODERN STATION BRING THE BOILER ROOM TO THE TURBINE

TO THE SUPERHEATER must be given the credit for much of the improvement in plant efficiency and flexibility made during the past few years. In this classification can be placed not only superheaters, but gas and steam reheaters (sometimes called resuperheaters) and desuperheaters. The first made high temperatures possible, the second made the reheat cycle possible and the third gave the flexibility and control necessary for practical operation in the high temperature region where metal strength and creep play such an important part.

On a unit basis the heat transfer boiler and superheater surface is about the same ranging from 5000 to 10,000 B.t.u. per sq. ft. per hr. as shown by Fig. 11. As far as boiler efficiency is concerned the effect of the two surfaces is the same and can be calculated from the general formula in the third preceding article.

The real value of superheat is not its effect on boiler efficiency but on the cycle and prime mover efficiency. Specific tests are mentioned in a previous article. It is difficult to generalize on this subject because the effect varies with several factors. In round figures, however, the use of 100 deg. F. of superheat instead of saturated steam will save from 20 to 30 per cent of the steam in a direct acting pump, from 18 to 20 per cent in a simple engine, from 14 to 16 per cent in a compound engine and from 10 to 12 per cent in a triple expansion engine. This saving is due largely to decreased cylinder condensation although small pressure and heat losses in piping valves and throttle play a part. This steam saving represents further operating economies in feedwater treatment and lighter boiler loads.

Cycle efficiency of course varies with the initial pressure and temperature and with the back pressure. Figure 1 shows graphically the variation in Rankine cycle efficiency for 1 in. abs. condenser pressure and different initial pressures and temperatures. The lowest solid line gives the efficiency when using saturated steam. Gains in efficiency due to superheat are shown by the 600, 800 and 1000 deg. F. solid lines while decreased efficiencies for higher back pressures are shown by the dotted lines. Increased initial temperatures with the higher back pressures will cause almost the same percentage increase as for 1 in. abs. condenser pressure as indicated by the solid lines[1].

Turbine performance is usually expressed in terms of Rankine cycle ratio, however, meaning the ratio of the actual machine performance to that of the ideal machine working under the same conditions[1]. The effect of pressure and temperature changes on efficiency is well shown by expansion lines A, B, C, D, and E. Efficiency ratios on the superheated region are from 10 to 20 per cent better than in the saturated region as shown by the closer approach of the expansion lines to the vertical.

The total heat drop of line A is about the same as line E but in the first case about 66 per cent of the line is in the superheated region and the Rankine ratio is about 81.5. In the second case conditions are reversed and only about 17 per cent is in the superheated region so the efficiency ratio drops to about 77 per cent. If line A extended from P to F^1 instead of from P to F the efficiency ratio would be only about 70 per cent. The 15 per cent increase is due to superheat[2]. Reheat has the same advantages. In this cycle, for instance MNOK, the superheated steam is expanded to about saturation and then re-superheated. Theoretically this could be done several times although the

[1]Turbine performance from the Mollier Diagram, Power Plant Engineering, Nov. 15, 1931, p. 1095.
[2]Draining Moisture from Turbines, Power Plant Engineering, October 1, 1931, p. 970.

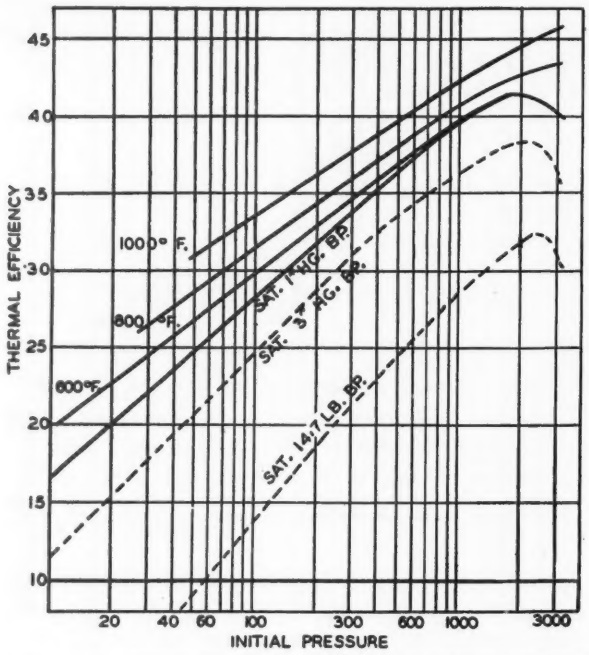


FIG. 1. RANKINE CYCLE EFFICIENCY IS INCREASED BY THE USE OF SUPERHEAT

decreasing pressure causes such an increase in steam volume that reheating more than once has not been commercially feasible.

Superheaters, not being cooled by water and limited to saturation temperature, are subjected to temperatures much higher than boiler surfaces. For higher temperatures alloy steels must be used, the following being the practice of one company[3]: For steam temperatures up to 825 deg. F. low carbon (0.15C) steel; for 850 deg. F., medium carbon (0.40C) steel; for 950 deg. F., low carbon steel calorized; for 1000 deg. F., medium carbon steel calorized; for 1050 to 1200 deg. F., 18 and 8 alloys. Metal temperatures range from 50 to 75 deg. F. higher than the steam temperatures. Alloy steels are expensive and the higher temperature installations sometimes use two section superheaters as at Burlington where the first bend is carbon steel and the second alloy. At Buzzard Point[4] the superheater is in two sections with a desuperheater between. This gives the close temperature control and allows carbon steel tubes to be used in both sections.

Temperature Control

Superheat control is necessary to protect the boiler piping and turbine from excessive temperatures. Ordinarily superheat from a convection unit increases with the rate of firing as shown by Fig. 3. This is just opposite from the radiant reheater and the two sometimes are placed in series to give a more constant superheat as shown by Fig. 4. Many other factors influence the final temperatures[5]. For instance gas firing gives lower temperatures than oil as shown by Fig. 4; low CO_2 gives higher superheat as shown by Fig. 5; location affects the superheat as shown by Fig. 6, the closer the convection superheater approaches the furnace the flatter the characteristic because a larger portion of the heat transfer is from radiation; baffle construction[6] is a factor as shown by Fig. 7.

Reheat requirements depend upon the method of turbine operation. As shown by Fig. 8, constant high pressure turbine exhaust gives a high temperature at light loads while variable exhaust pressure gives low temperatures at light loads. This can be compensated for by a suitable selection of superheater surfaces as shown by Fig. 9, although it is sometimes accomplished by desuperheaters or steam reheaters[5].

There are wide variations in practice to meet local conditions of operation, equipment, space and fuel. The radiant superheater is of considerable importance to industrial establishments because of the ease with which it can be added, usually to the back wall, of exciting furnaces to be used as a superheat booster as in Fig. 14.

Flooding

Formerly it was considered necessary to flood superheaters when starting up a boiler. That is, the superheater elements were filled with water so that they would not overheat. This is no longer considered necessary, sufficient justification for the change being shown by the results of tests shown by Figs. 12 and 13. The dry method of operation actually shows lower metal temperatures than flooding[7]. The small steam flow from the drain of course gives some cooling after the pressure starts to rise. To give protection when the load is suddenly dropped the superheater safety valve should be set to pop at a lower pressure than the valves on the saturated steam drum[8].

[3]N.E.L.A. Proceedings, 1931, p. 787.
[4]Buzzard Point Plant Supplies Power to Capitol, by H. G. Thielscher, Power Plant Engineering, Nov. 1933, p. 478.
[5]Reheat Practice, Operation and Control, Power Plant Engineering, August 1, 1932, p. 588; October 1932, p. 703 and December 1932, p. 806.
[6]Power Plant Engineering, Feb. 1, 1930, p. 169.
[7]N.E.L.A. Publication No 241, January 1933, p. 6.
[8]N.E.L.A. Proceedings 1929, p. 1304.

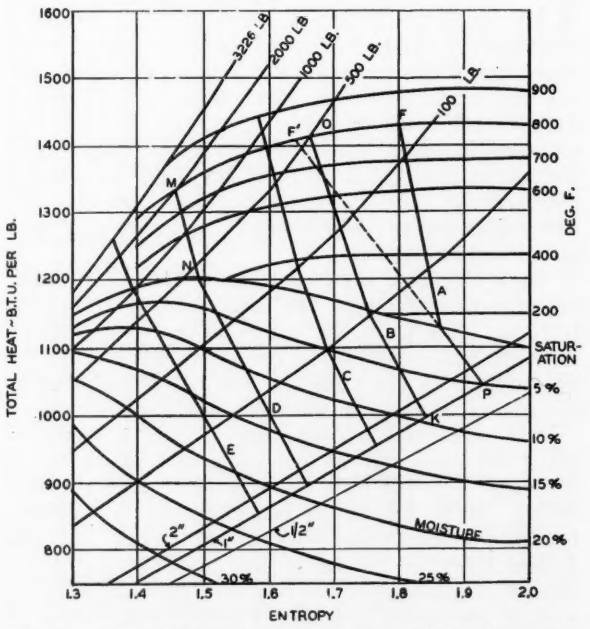


FIG. 2. APPROXIMATE EXPANSION LINES OF TURBINES SHOWING THE EFFECT OF SUPERHEAT ON TURBINE EFFICIENCY. THE CLOSER THE LINE APPROACHES THE VERTICAL THE HIGHER THE EFFICIENCY

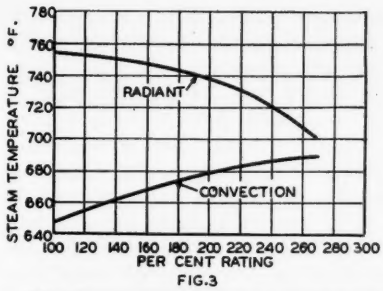


FIG.3

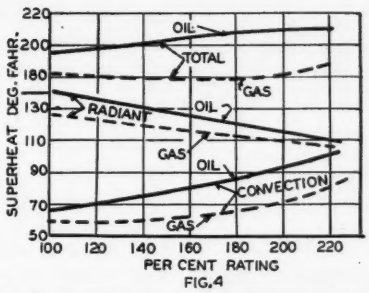


FIG.4

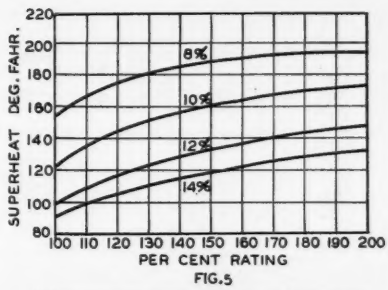


FIG.5

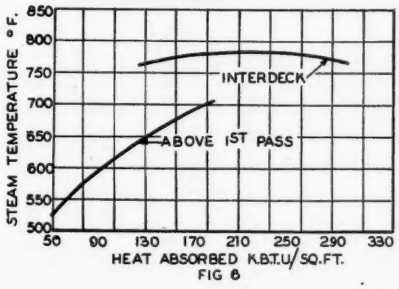


FIG 6

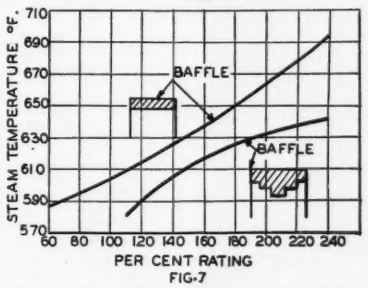


FIG.7

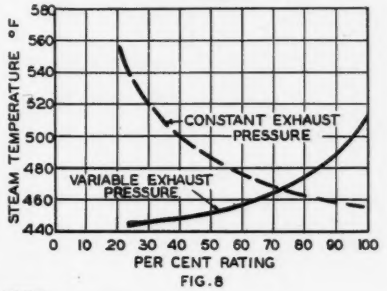


FIG.8

DATA SHOWING SUPERHEATER PERFORMANCE

Fig. 3. Radiant and convection superheaters have different characteristics

Fig. 4. By combining radiant and convection elements in series an almost constant superheat can be obtained over the entire load range

Fig. 5. CO_2 content changes the superheat and has a slight effect on the shape of the characteristic curve

Fig. 6. In this boiler the convection superheat characteristic was changed by adding additional surface and moving the elements 8 tubes nearer the furnace

Fig. 7. Convection superheat characteristic changed by baffling the gas flow

Fig. 8. Temperature of steam to the reheater turbine exhaust depends upon the method of turbine operation

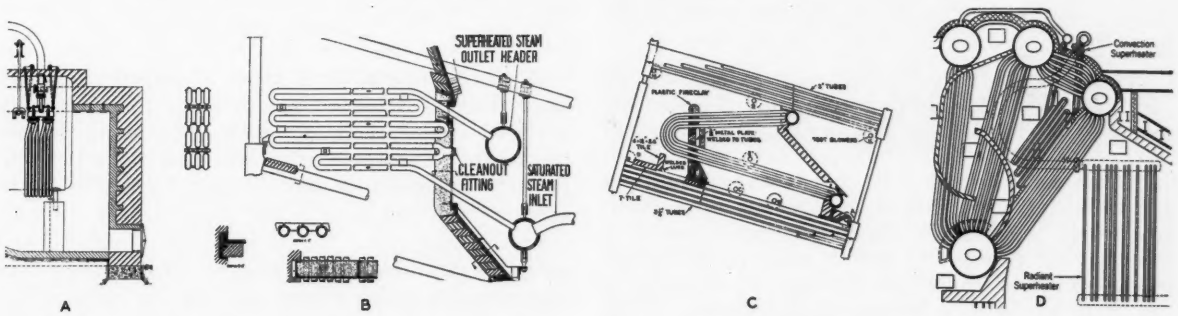


FIG. 10. REPRESENTATIVE TYPES AND LOCATIONS AND SUPPORTS OF SEVERAL TYPES OF SUPERHEATERS

There is lack of understanding on the part of some designing and operating engineers as to the theory of setting superheater safety valves. In superheaters, located in a zone of high gas temperature and high heat transfer rates, it is necessary to maintain a certain velocity through the superheater tubes in order to protect them from burning, due to an excessive difference between the temperature of the metal in the tubes and of the steam within them. Such velocities result in building up a pressure drop through the superheater.

Were the superheater and saturated safety valves set to blow at the same pressure, the pressure drop existing between the boiler drum and the superheater outlet would cause the saturated valve to blow first and the superheater would have little or no flow of steam through the tubes for their protection. It is necessary to get the superheater safety valves at a pressure below that of the saturated valves in order that the superheater valves may blow first, insuring a flow of steam through the superheater at all times.

Other conditions which make lower superheater

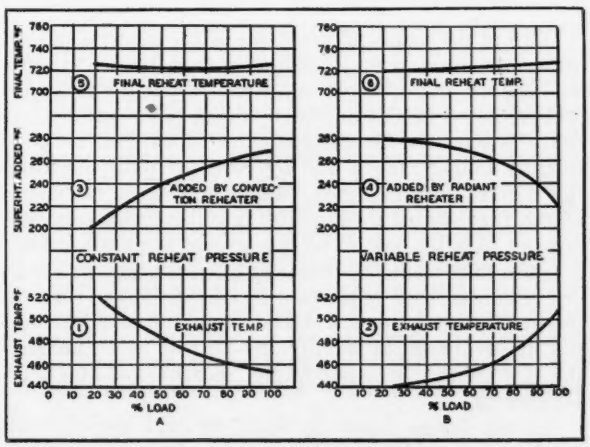


FIG. 9. (A) APPLICATION OF A CONVECTION REHEATER TO A TURBINE INSTALLATION OPERATING AT CONSTANT EXHAUST PRESSURE AND (B) A RADIANT SUPERHEATER TO A TURBINE INSTALLATION OPERATING AT VARIABLE EXHAUST PRESSURE TO GIVE A CONSTANT REHEAT

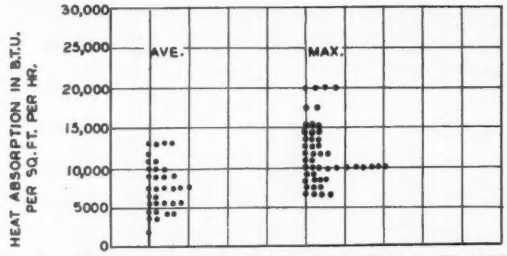


FIG. 11. AVERAGE AND MAXIMUM HEAT TRANSFER RATES FOR SUPERHEATERS INSTALLED IN A NUMBER OF PLANTS

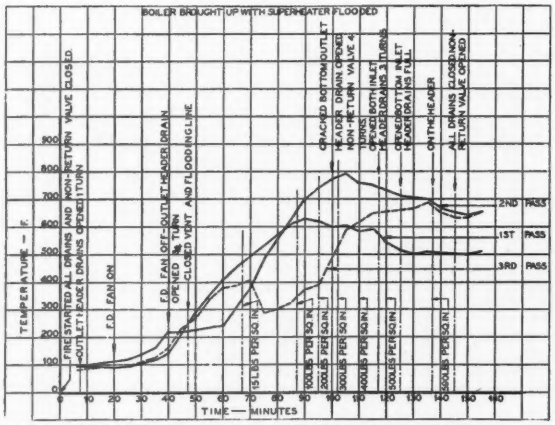


FIG. 12. SUPERHEATER TUBE TEMPERATURES WITH FLOODED TUBES

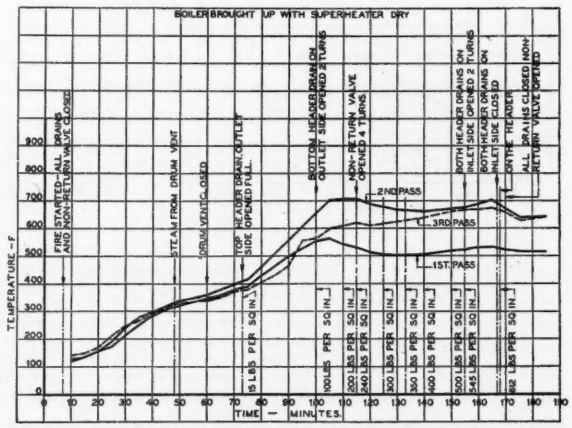


FIG. 13. SUPERHEATER TUBE TEMPERATURES WITHOUT FLOODED TUBES

safety valve setting necessary occur in installations where a number of boilers are connected to a common steam main. Under such conditions, if a saturated valve were to blow before a superheater valve, all of the steam generated by the boiler blowing would not pass through the superheater and the amount of superheat would increase. On slacking the fire in order to cause the safety valve to shut, a smaller and smaller amount of steam would pass through the superheater and there would be a corresponding increase in the superheat.

Steam Distribution

Two of the principal problems of the modern superheater are even steam distribution from intake to outlet headers and durable supports. Even distribution is obtained by header design, by using fewer and longer tubes in parallel, by using smaller tubes with proportionally longer length and by cores or plugs limiting the flow to certain tubes. Multiple loop superheaters of the type shown by Fig. 10B may be either counterflow or parallel flow. The one shown is parallel flow Counter flow would be obtained by reversing the headers. Counter flow requires only about 85 per cent of the surface and gives a flatter characteristic but has the disadvantage of having the hottest tubes nearest the fire with correspondingly increased danger of overheating.

In the design shown by Fig. 10B each loop is held in place by welded-on spacers with the outer end of the lowest loop resting on a header bracket. Ends of the tubes are expanded into the headers which are protected from the hottest gases. In some designs the headers are placed entirely outside of the boiler setting. In the design shown by Fig. 10C a single cross wall consisting of cast-iron saddles across the width of the boiler supports the tubes.

The saddle rests on the top row of tubes and is held in place by U-bolts around them. Calorized flame plates bolted to the saddles extend down through four rows of boiler tubes to distribute the weight. Calorized flame plates are used in the bottom and top loops to keep the tubes properly spaced and the top and bottom of each inner loop to act as a reinforcing for the high temperature cement used to cover all metal surfaces exposed to the flame.

In h.r.t. boilers three types of superheaters are used. One of these shown by Fig. 10A is simply strapped around the boiler without changing the setting or arch. So located, it absorbs both convection and radiant heat to give a desirable flat characteristic. In another type the superheater elements are dropped from the rear arch. This is also easily installed and possesses desirable characteristics. Still a third type uses elements which extend into the fire tubes of the boiler. Other superheater positions are shown by boiler illustrations in other articles of this issue.

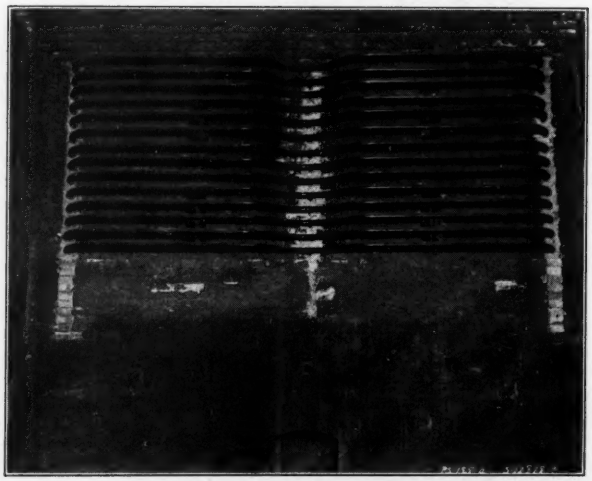
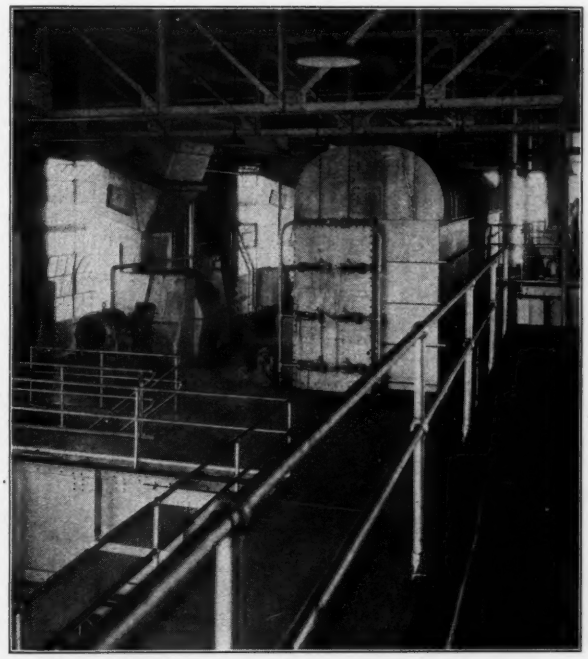


FIG. 14. SHOWING HOW LITTLE BRICKWORK MUST BE REMOVED TO INSTALL A RADIANT SUPERHEATER TO ACT AS A BOOSTER IN AN EXISTING BOILER

Economizers *and* Air Heaters *for* Higher Efficiency

Heat Recovery By these Methods More Economical Than Additional Boiler Surface and Increased Flexibility Helps Meet Modern Needs in the Modern Way



BOTH ECONOMIZERS and air heaters are, fundamentally, waste heat recovery equipment originally intended to recover heat which the boiler surface proper could not absorb for practical or economic reasons. Besides being expensive, boiler surface is limited, as far as efficiency is concerned, by being able to cool the gases of combustion only to within 50 or 100 deg. F. of the saturated steam temperature in the drum. The original function of the economizer was to recover as much as possible of this waste heat.

As the economizer developed, however, its utility and advantages became well established and its lower unit cost enabled it slowly to progress to its ultimate development, the steaming economizer which heats the feedwater up to the steaming point. Perhaps it is not proper to say the economizer encroached on the boiler of its own free will. It was forced upward in the temperature scale in order to justify its utilization because the regenerative feedwater heater was gradually ousting the economizer from the low temperature range. As the economizer was forced up the scale, the air heater came in and took over the function originally performed by the economizer, the recovery of low temperature heat.

Heat Transfer

On a unit basis the heat transfer of economizers compares favorably with that of boilers. Although most of this heat transfer is by convection. The flexibility of economizer construction is such that distribution of heating surface and proportioning of gas flow can be carefully controlled to give best results. Conservative heat transfer rates for a small economizer are shown by Fig. 1. In practice, in large modern units these rates are often exceeded by 200 or 300 per cent[1]. On a square foot basis the heat actually transferred depends upon the temperature difference between the water and gas and in practice averages somewhat over 1000 B.t.u. per sq. ft. per hr. as shown by Fig. 3.

Selection and design of the economizer is of course beyond the scope of this article. For this information the reader is referred to the article "How Do You Select Your Economizer?" by Theodore Maynz. This article, which appeared on page 1214 of the Nov. 1, 1930 issue of Power Plant Engineering, discusses the design and selection of economizers from both the engineering and financial side. Figure 4 taken from that article shows the relation of surface to gas flow, temperatures and heat transfer.

Increase in efficiency due to an economizer in any particular case can be calculated from the general boiler formula in a previous article if the temperature of the water entering and leaving the economizer is known or assumed. It is usual to design economizers on the counterflow principle so that the incoming cold water meets the outgoing and comparatively cool gases while the hot water leaving the economizer encounters the hot gases just entering. This gives the greatest mean temperature difference and consequently the greatest total heat transfer per unit of surface between the two fluids and at the same time allows the gases to be cooled lower and the water to be heated higher than would otherwise be the case with the same surface. The temperature changes of the water and gas depend upon their relative weights, and are related by the equation[2]

$$(t_2 - t_1) \div (T_1 - T_2) = C\ (W \div w)$$

where t_1 and t_2 are the initial and final temperatures of the water, T_1 and T_2 the initial and final tempera-

[1] See Heat Transfer in Air Heaters and Economizers, by Henry Kreisinger, A.S.M.E. Annual Meeting December 1933. (See short abstract in A.S.M.E. report in this issue.) Also see p. 1227 N.E.L.A. Proceedings 1926.

[2] How to Check Your Economizer Guarantees, by B. M. Thornton, Power Plant Engineering, Feb. 1, 1929, p. 171.

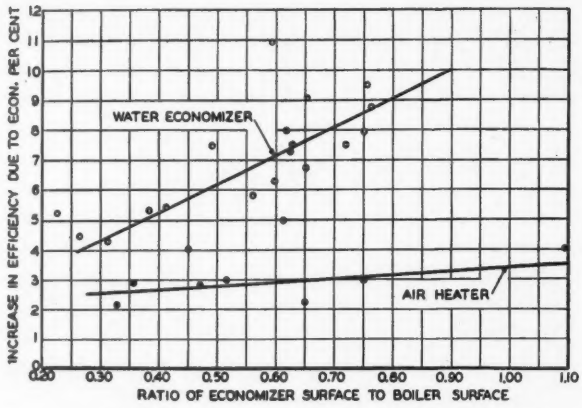


FIG. 1. HEAT RECOVERY AS A PERCENTAGE BEARS A DEFINITE RELATION TO THE SURFACE INSTALLED

tures of the flue gas, w is the weight of water and W the weight of flue gas passing through the economizer per hour and C is the mean specific heat of flue gas which for usual calculations may be taken as constant at 0.245 B.t.u. per lb. per deg. F. Water flow may be measured by meter, either as water or steam, temperatures may be measured, so W is easily calculated.

In design problems the gas flow must be calculated from the assumed fuel and operating conditions. In practice, air admitted to the furnace bears to the gases of combustion[3] a ratio of about 12 to 13 so that the weight of gases available from a given quantity of coal

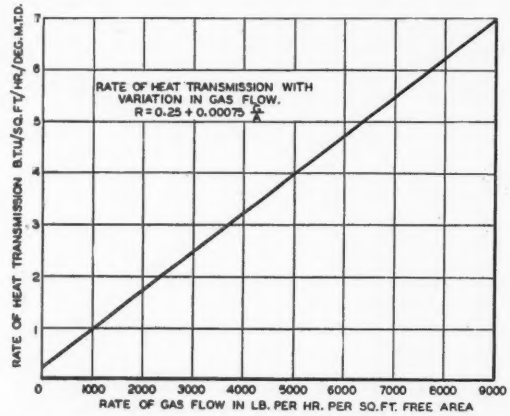

RATE OF HEAT TRANSMISSION B.T.U./SQ.FT./HR./DEG.M.T.D.
RATE OF HEAT TRANSMISSION WITH VARIATION IN GAS FLOW.
R = 0.25 + 0.00075 W/A
RATE OF GAS FLOW IN LB. PER HR. PER SQ.FT. FREE AREA

FIG. 2. HEAT TRANSFER TESTS OF A STEEL TUBE ECONOMIZER

burned is a simple proportion. When economizers depart from estimated performance, the cause is usually due to a variation in the ratio of weights. This may be due to defective fuel burning equipment giving a lower CO_2 than anticipated or more frequently due to the use of a fuel for which the design was not intended. Change in CO_2 changes the weight of gas flow and has the same effect on temperature as mentioned in connection with superheaters.

Originally cast iron was the usual economizer metal due to its low cost and resistance to corrosion. Economizers cooling the gases to low temperatures are often wet on the outside due to condensation from the flue gases and this favors external corrosion, especially with high sulphur coal. More important perhaps is the matter of internal corrosion due largely to the fact that any oxygen in the feedwater is liberated in this section. Oxygen is not of course necessary for corrosion to take place, however, as has been discovered by experience. The matter of pH or acidity of the boiler water plays an important part[4]. In spite of the fact that pH 7.5 is the neutral point boiler water has to be decidedly alkaline, pH 9.5, to remove the danger of corrosion.

With increasing pressures the limited strength of cast iron precluded its use and steel tubes were substituted although these steel tubes are sometimes coated with lead or covered with shrunk-on cast-iron rings as shown by Fig. 5B. In some cases, particularly in reconstruction where old low pressure economizers are still serviceable, a low and high pressure boiler feed pump has been used. Although complicating the design by an extra pump this scheme allows the use of a low pressure economizer with a high pressure boiler. The flexibility of the modern economizer is well shown by Fig. 6 added to the uptake of an h.r.t. boiler.

Details of the designs of several manufacturers are shown by Fig. 5. Gas leakage around the tubes is prevented by the packed stuffing box, shown by Fig. 5A. In order to clean or replace the tubes, the ends are connected together with return bends using flanged connections. A similar flanged connection is used on the header. Another design using handholes and plates is shown by Fig. 5B while the complete end assembly of still a third design is shown by Fig. 5C.

This design uses finned tubes electrically welded at the top and bottom to give additional heat absorbing surface. The unit shown has flanged return bends with ball joints as furnished for the higher pressures where accessibility for cleaning is necessary. Another type, for lower pressures, consists of two straight tubes connected at one end by a welded return bend and with each U tube connected to a header. When feedwater conditions are good and no cleaning is necessary the elements are sometimes made use of as a continuous tube with return bends welded in and with the inlet and outlet expanded into headers. Figure 5D shows the extended surface elements of Fig. 5B assembled as a unit with sootblower elements in place.

Air Heaters

Although air heaters recover heat from waste gases the same as economizers, their effect on efficiency is

[4]Placing pH in Corrosion and Water Treatment, by S. E. Tray, Power Plant Engineering, June 15, 1932, p. 487.

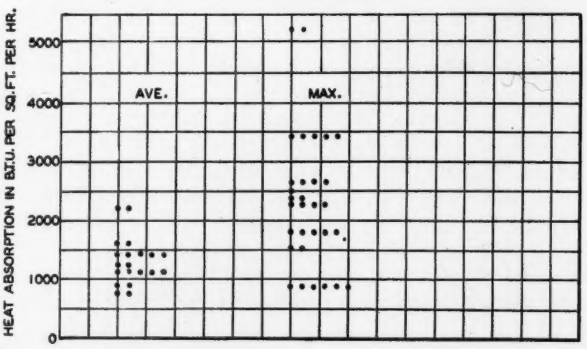


FIG. 3. TYPICAL PRACTICE IN HEAT TRANSFER THROUGH ECONOMIZER SURFACES

[3]The Thermal Balance of Air Heaters of Large Steam Generators, by M. W. Travers, Engineering of London, Aug. 5, 1932, p. 161. Also same title by W. F. Harlow, Sept. 23, 1932, p. 356.

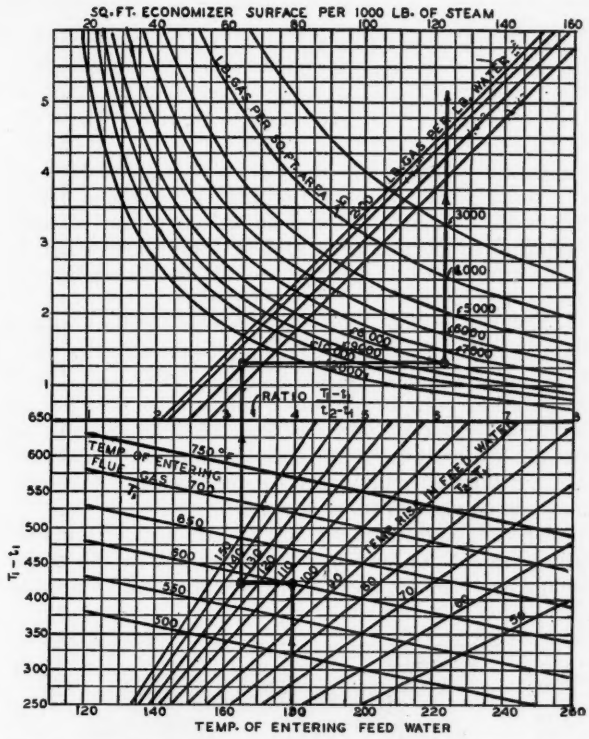


FIG. 4. CHART FOR CALCULATING ECONOMIZER SURFACE

entirely different. Measured in B.t.u. the heat recovered is small but the effect of this heat recirculated back through the furnace materially improves the capacity of the furnace as shown by the first article of this issue. The importance of air heated to the highest possible temperature is reflected in the rapid increase of air temperatures as shown by Fig. 8.

Three general types are in wide use. The type shown by Fig. 7A consists of plates with the air on one side and gas on the other, while the type shown by Fig. 7B consists of tubes through which the flue gases travel, the air properly divided by suitable baffles traveling over the outside of the tubes. A third type shown by Fig. 7C is usually known as the regenerative type, the revolving metal elements being alternately heated by the gas stream and then cooled by the incoming air.

Although usually built counterflow like economizers, the heat transfer coefficient is only about 20 per cent that of economizers because of the lower heat transfer from metal to air. Heat[1] is transferred from the hot gas by convection into the gas film; from the gas film it is transferred by conduction through the layer of soot and ash, and through metal wall into the air film; from the air film it is transferred by convection into the moving body of air.

The rate of heat transfer by convection in the gas and air stream in an air heater varies with the weight of gas and air or water passing over the surfaces. The temperature is low and the range of temperature variation is small so that the effect of temperature on the rate of heat transfer is negligible. The rate of heat transfer by conduction varies directly with the conductivity of the material forming various layers, films and metal walls and inversely as their thickness.

Conductivity of the metal wall is high compared to the conductivity of the films and the layer of scale and particularly the conductivity of the layer of soot and ash on the gas side of the metal plate. This layer of soot and ash is a factor which greatly affects the heat transfer. Inasmuch as the layer may be of varying thickness and density in different installations, the rate of heat transfer varies appreciably, and makes close prediction of the efficiency of the apparatus difficult.

In the calculation of the performance of air heat-

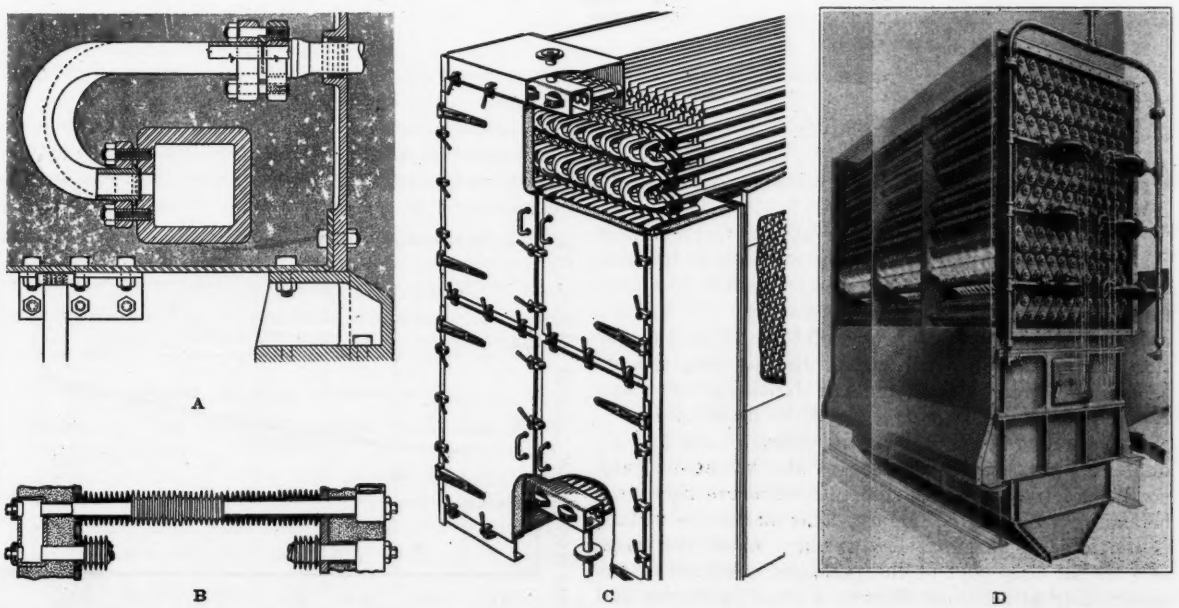


FIG. 5. A. DETAILS OF ECONOMIZER RETURN BEND AND HEADER CONNECTIONS. B. ANOTHER ECONOMIZER DESIGN. C. COMPLETE ECONOMIZER ASSEMBLY. D. PHANTOM VIEW OF AN ECONOMIZER FROM THE SOOT BLOWER END

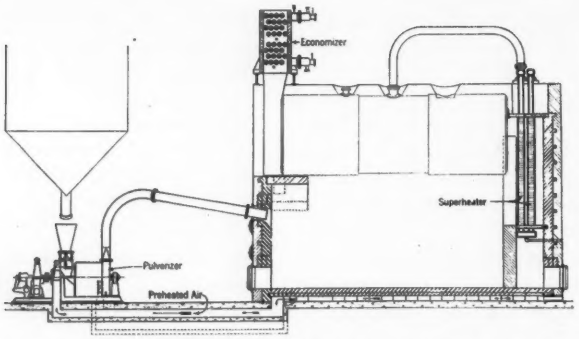


FIG. 6. THE MODERN ECONOMIZER CAN BE LOCATED PRACTICALLY ANYWHERE

ers the rate of heat transfer is taken to vary with the weight of gas and air per square foot of cross section called the mass flow and is adjusted to the probable cleanliness of the heating surfaces. Such adjustment can be made intelligently only from test results. The weight of air is usually considerably smaller than the weight of gases, particularly at low rating. Therefore, the weight used in the calculation is the average of the weight of gases and air. In certain types of air heater the air passages are made smaller in order to increase the air mass flow.

Recirculation of part of the heated air increases the mass flow and thereby the rate of the heat transfer and may offset the detriment due to smaller temperature difference. It has the advantage of greatly reducing the condensation of water vapor on the gas side surfaces at the entrance of the air into the air heater and thereby reducing the deposit of soot and ash and also avoiding corrosion of the metal plates.

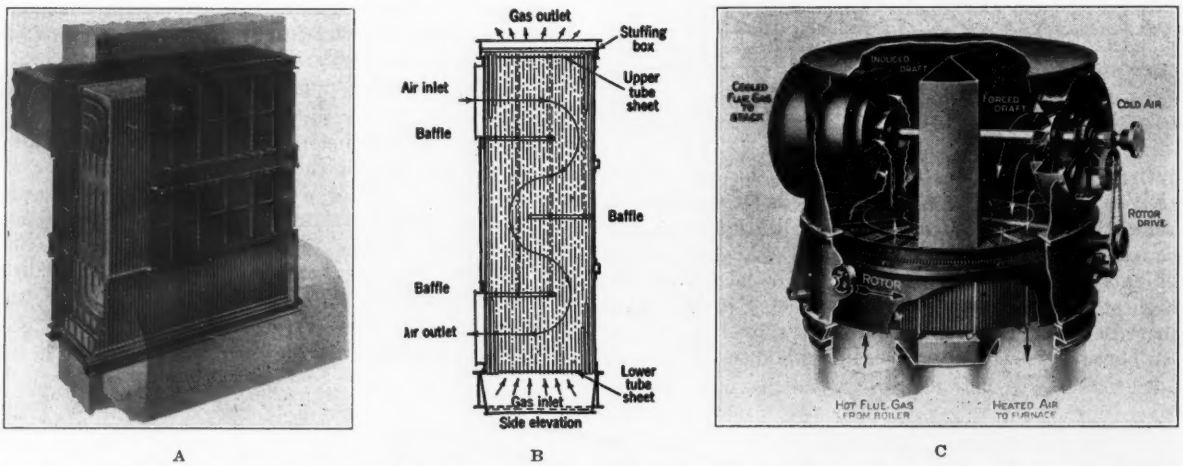
On account of suitable arrangements of gas and air ducts, all the heating surface of the air heater may not be effective and in the calculations correction must be made for ineffective surface. This ineffective surface is particularly large at low ratings, when the gases and air fill only part of the passages. Because of the uncertainty of all these factors, a good judgment and comparison with test results are of greater importance than exact formulas.

Heat Transfer

Figure 9 shows probable heat transfer values for air heaters[1] although as pointed out above the performance of the unit depends upon so many factors that it is difficult to give definite figures. When all of the air of combustion is heated by flue gases the temperature rise of the air bears a definite relation to the temperature drop of the gas. In an ideal plant this ratio[3] is about 1.1 but in actual plants it may range up to 1.5 and even higher if high excess air is used or infiltration through the boiler setting is considerable.

One factor tending to increase this ratio escaped detection until recently. For practical purposes combustion is assumed to be complete after the boiler surface has been swept over. Combustion precedes at low temperatures, however, and takes place even in the air heater thus giving a higher temperature rise to the air than accounted for by the specific heat, flow and temperature drop of the gas.

One has only to realize that 1 per cent of the fuel bill is saved for every 30 deg. F. fall in the stack temperature to appreciate the bearing which the final temperature has on the financial return from the plant and that a very considerable expenditure may be shown to be justified for any means for reducing this temperature to certain limits. What this limit is



A B C

FIG. 7. THREE TYPES OF AIR HEATERS WIDELY USED. A. PLAIN TYPE. B. TUBE TYPE. C. REGENERATIVE TYPE

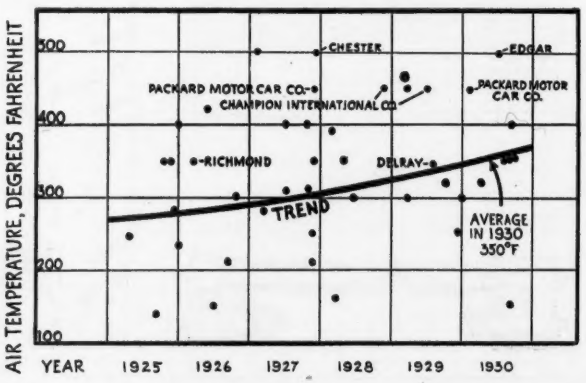


FIG. 8. INCREASE IN PREHEATED AIR TEMPERATURE FOR UNDERFEED STOKERS

depends on the initial air heater cost, the maintenance charges on the same, the price of fuel and the load factor of the plant but there are very few power stations where it does not pay to reduce the gases to 250 deg. F.

All parts of the modern boiler work together to bring the flue gases to the lowest economic temperature. The function of each element in a typical installation[5] is shown by Fig. 10, the lower heat transfer in the convection sections are apparent at a glance. Usually the air heater follows the economizer although that is not always the case. For instance at Huntley No. 2 the air heater is in parallel with the economizer. It receives a portion of the hot flue gas and heats about 10 per cent of the air for combustion, that is sufficient 400 deg. F. air for mill drying.

Another somewhat similar installation was made at Hell Gate for the same purpose although structural

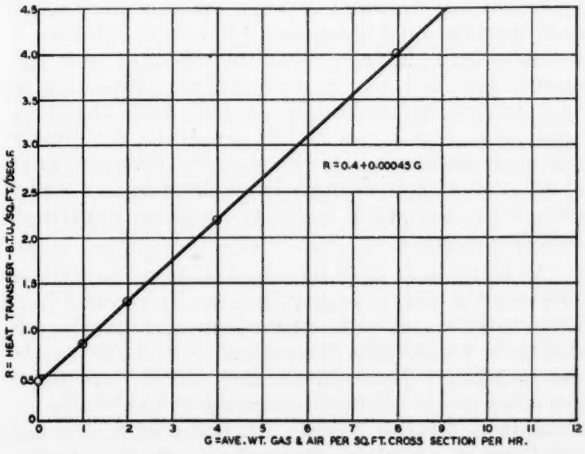


FIG. 9. UNIT HEAT TRANSFER IN AIR HEATERS[1]

limitations placed by the old building was a factor in the decision. In this installation a 26,136 sq. ft. economizer and a 11,840 sq. ft. air heater are in parallel, the latter supplying heated air to the pulverizer. The temperature of the air is under control of dampers normally operated so that the gas outlet temperature of the economizer and air heater is the same. Under abnormal conditions, as for instance wet coal, the dampers can be regulated to give the necessary air temperature.

In cases of this kind the economizer and air heater must be considered as a single unit, an extension perhaps of the so-called supermiser[6] developed in England a few years ago although in that piece of equipment the combination was affected by means of concentric tubes.

Two other special air heater installations warrant mention. One of these at Stanton Station[7] uses air heaters in the flue gas stream and economizers in the hot air stream. The purpose of this arrangement was to eliminate external economizer corrosion and at the same time serve as a tempering device for the air to the stokers. It has not been used since probably

[5]A.S.M.E. Transactions, FPS-50-70, 1928.
[6]Economizer and Air Heater Combined by British Engineer. Power Plant Engineering, Apr. 15, 1929, p. 466.
[7]Power Plant Engineering, Sept. 1, 1927, p. 916.
[8]Power Plant Engineering, May 1, 1931, p. 486.

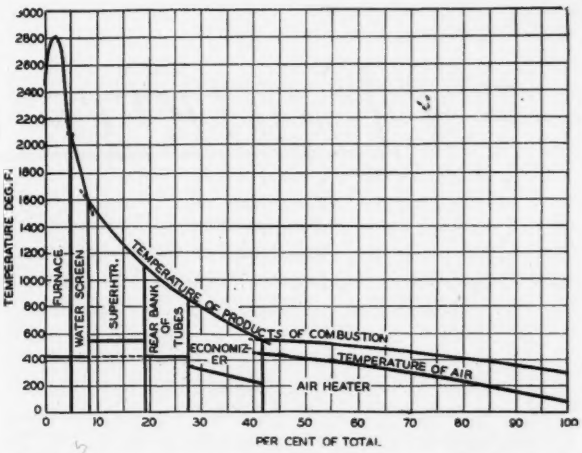


FIG. 10. PROPORTIONING OF THE VARIOUS SURFACES IN A STEAM GENERATOR IS SUBJECT TO CONSIDERABLE VARIATION TO MEET LOCAL CONDITIONS

because corrosion troubles and use of preheated air are now better understood.

The second special installation is at Bremo Station[8] where the high single pass boiler made hot air ducts inadvisable. In this station diphenyl is used as a heat transfer medium. It is heated in an economizer, pumped down to heat exchanger at the burner and there gives up its heat to air entering the furnace.

Research on Heating at Mellon Institute

ACCORDING to an announcement by Dr. Edward R. Weidlein, Director, Mellon Institute of Industrial Research, Pittsburgh, Pa., the Multiple Industrial Fellowship on heating, sustained in the Institute since 1929 by the National Radiator Corp., Johnstown, Pa., is continuing actively a number of investigations of interest to heating specialists as well as users of heating equipment. In addition, the Fellowship, whose incumbents are Dr. J. L. Young and A. C. Jephson, has been lately according more and more research attention to certain problems in air conditioning and also to the development of new products.

After a thorough investigation of ferrous and nonferrous heating units, the Fellowship has found that cast iron is the most suitable metal for the construction of finned convectors. Results of work on impregnated wood foundry patterns, now fully patented, are likewise of interest to heating industrialists; this development is of broad application in the foundry field. During the past three years the Fellowship has been engaged in the development of corrosion-resistant materials especially for use in the petroleum and other process industries, conducting this investigation in collaboration with the research department of the National Radiator Corp. During the course of these studies the findings of the International Nickel Co., respecting the place and utility of "Ni-Resist" have been corroborated. The Fellowship has also had a productive part in the design of National condensing sections, for which broad patent protection has been secured.

Features of Modern Fur

The question of Furnace Design is one which involves a careful study of the entire steam generating unit, as the type of furnace used in any particular installation must be designed to fulfill all the operating conditions of that installation.

FURNACE DESIGN for large and medium size power boilers has undergone rapid and remarkable development during recent years. It is, of course, impossible to separate developments in furnace design from those in stoker and pulverized coal practice and boiler practice in general but if these relations are kept in mind, it is no exaggeration to say that developments in furnace design have had as far reaching an effect on general power plant development as any other single factor in the field. And this is not surprising; the furnace is unquestionably one of the important parts of the power generating system, indeed there is no other apparatus in the power plant upon which so much depends as the boiler furnace. For it is in the furnace that the major item in the power cycle is introduced, i.e., fuel, and any savings effected here or any unnecessary loss incurred here is multiplied by the operating limitations in other parts of the system.

Nearly all of this development in furnace practice has come about since 1920 or perhaps the last decade. In that period pulverized fuel firing advanced from a purely experimental stage to a universally recognized system of which the reliability and high efficiency has been shown by tests and operating results in many plants. The efforts toward improvement in pulverized fuel systems of firing have extended to every part of the system and have had a direct bearing on stoker design and practice. Indeed, it is questionable whether the rapid strides made in stoker design and performance would have been made if stoker practice had not been confronted with the competitive influence of the pulverized fuel systems.

In this period, the completely water cooled furnace as we understand it today came into existence and opened up new avenues towards more effective operation and decreased maintenance. This in a way was a reversion to the principle of the old internally fired boiler but was the result of a much more scientific attitude towards combustion phenomenon in furnaces, a better understanding of what was required in a furnace. There is today a much better appreciation of such factors as combustion volume, heat liberation, combustion period, etc. In this period too, the general use of preheated air for combustion purposes came about and with it changes in economizer and boiler surface practice, all of which had a direct bearing on furnace design.

As boiler sizes have increased so have the sizes of furnaces but not in proportion, for the rates of heat liberation per unit volume have increased greatly; not nearly to the possible theoretical limits however. In this particular phase of furnace design, the paper delivered at the Second International Conference on Bituminous Coal in 1928 by Dr. Rosin, of the Mining Academy at Freidberg, Germany, is of great importance. This paper,[1] which has had altogether too little attention in this country, attacked the problem of combustion of pulverized coal from a purely theoretical standpoint and on the basis of such factors, as fineness of pulverization, velocity, burning time, etc., he determined the theoretical limits of heat liberation in furnaces under various conditions. The results of his conclusion are quite astonishing. Theoretically, the

[1]Thermodynamics of Powdered Coal Combustion. Power Plant Engineering, Mar. 15, 1930, p. 326.

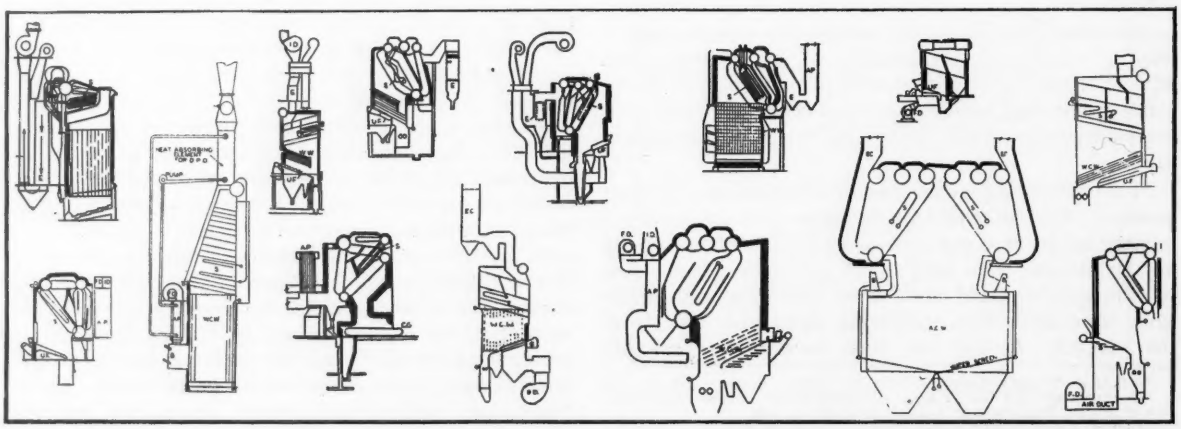


FIG. 1. DIAGRAMS SHOWING VARIOUS SHAPES OF FURNACES USED IN MODERN INDUSTRIAL AND UTILITY PRACTICE

r nace Practice

rate of heat liberation can be increased to enormous limits by reducing the size of coal particles. Dr. Rosin's calculations[1] show that the rate of heat liberation varies from 12,600 B.t.u. per cu. ft. per hr. for particles 200 microns long to 2,770,000 B.t.u. per cu. ft. per hr. for 10 micron particles! It is shown, however, that this high theoretical limit cannot be reached because of other factors, temperature and turbulence, etc. The mere fact, however, that the theoretic limit of heat liberation is so enormously greater than that obtained in practice should encourage designers to further research and investigation. At the present time with furnaces of modern design using pulverized fuel or stokers, heat liberation varies anywhere from 10,000 B.t.u. per cu. ft. per hr. to as high as 65,000 B.t.u. per cu. ft. per hr.

In the accompanying table (Table II) we present furnace data on a number of central stations built since 1927. It will be noted here that the average heat liberation in the 31 furnaces on which data is presented is in the order of 30,000 B.t.u. per cu. ft. of furnace volume per hour. While this figure is far removed from the high theoretical values of Dr. Rosin, it represents modern practice and is considerably greater than it was before, say, 1920. These figures are all for central station plants. In small plants of the industrial type without water walls, a value of around 15,000 B.t.u. per cu. ft. is common.

The design of a furnace should take into account first, of the type, quality and quantity of fuel it is to burn; second, the duration of the peak load and the type of combustion equipment, that is, grates, stoker, pulverized fuel or oil burner. If burners are to be used, it is necessary to know the character of the flame, whether long or short, its angle and the temperature zones. This of course will depend upon the type of burner used. Sufficient time is required for combustion; for pulverized coal, this means about 15 ft. of travel. With tangential firing and with turbulence this travel is obtained in the minimum of cubical furnace content. The length of time required for burning of course will depend upon the fuel, its size, air fuel ratios and their mixture temperatures.

The shape of the furnace is an important factor in its design. Although it is determined largely by the shape and type of the boiler, still, the designer has considerable margin in determining its shape. In general, with pulverized coal, for vertical and horizontal firing, a wide and short furnace is better than a long and narrow one. The wide furnace permits more burners and thereby makes possible a better distribution of coal and air throughout the furnace. This consideration applies equally well to gas and oil fired furnaces and to a certain extent for stokers. In the case of stokers, the shape of the furnace is to a large extent governed by the type and shape of stoker.

With respect to this consideration of furnace shape it is of interest to note the great variety of furnace shapes in use in modern installations. In the accompanying diagram a number of modern boiler installations are shown. It is quite evident from these elevations that the shapes of furnaces vary considerably. In plan modern furnaces present greater uniformity. Table I shows the length, width, height, together with total furnace volume of some 40 modern boiler installations. While these figures perhaps have little design value it is interesting to note that the average of these 40 furnaces comes out about 24 ft. square. The average of the heights is approximately 30 ft. These figures are of course for the larger size furnaces used in central stations. Setting heights for large and medium size boilers using stokers are given in Table III.

In the design of a furnace the starting point is usually the boiler heating surface and the rating at which the unit is to be operated. This gives the total heat input necessary in the furnace. Knowing the heating surface, the rating of operation and assuming a figure for overall efficiency, the total heat release per hr. is given by the following formula:

$$\frac{(\text{H S} \div 10) \times 33{,}500 \times \%\ \text{rating}}{\%\ \text{Efficiency}}$$

This gives the total heat release per hr. in B.t.u. for the entire furnace. To obtain a figure for the furnace volume, divide this figure by the permissible heat release per cu. ft. per hr. The permissible heat release

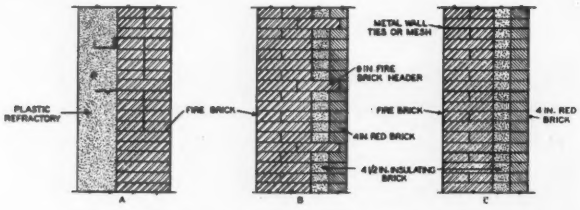


FIG 2. TYPES OF SOLID REFRACTORY WALLS

for any given furnace depends upon several factors, the fusing temperature of the ash, the type of refractory or furnace walls, etc. H. W. Brooks gives the following heat liberation values for pulverized coal furnaces, though he states that higher values have been successfully used in some cases. His maximum of 30,000 B.t.u. per cu. ft. is considerably above the average for the 12 pulverized coal installations on which heat release data is given in Table II. The average for the 12 installations listed is 25,055.

For stoker fired furnaces the heat release values are higher. This is due to the relatively high velocity between the air and fuel. As shown in Table II in the case of the Kidder Station a value of 64,000 B.t.u. is reached. In small plant design it is well to stay between 15,000 and 25,000 B.t.u. per cu. ft.; 15,000 for lignite, 25,000 for anthracite. With gas and oil, values between 25,000 and 50,000 represent modern practice.

In any instance the values used will be dictated largely by the type of furnace walls used. There are three general types in use: (1) solid refractory; (2) air cooled refractory, and (3) water cooled, either totally or partially. Which type to select depends largely upon the rate at which heat is to be released, i.e., upon the rating at which the unit is to be operated. Solid refractory walls are suitable and the most economical for furnaces which operate below 150 per cent rating. Between 150 and 250 per cent, some form of air cooled wall is advisable and in general above 250 per cent the water cooled wall is the most economical.

Solid Refractory Walls

The principal requirement of a solid refractory wall is to withstand the furnace temperature without structural failure. The actual construction may take any of a large variety of forms. Usually, however, such walls consist of a lining of refractory over a heat insulating brick of some sort and with an outside facing of ordinary brick, or second grade fire brick. Instead of an inside lining of refractory brick, a jointless monolithic wall lining may be provided by the use of plastic fireclays. In laying brick for furnace walls, the brick should be laid up as closely as possible, at least insofar as horizontal joints are concerned. In

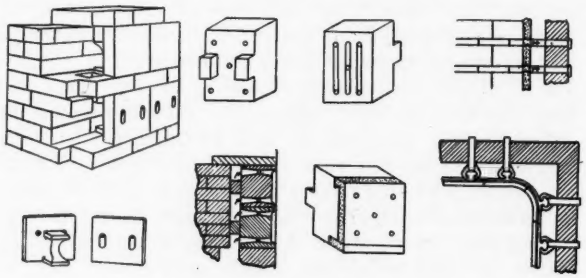


FIG. 3. TYPES OF AIR COOLED SELF SUPPORTING FURNACE BLOCKS

advanced modern practice, the top of each course is rubbed to a substantially plane surface with silicon carbide brick or other abrasive before the next course is placed. Furthermore, the quantity of binder material between the courses should be limited to that required to fill unavoidable depressions in the adjacent surfaces. A thin "Slurry" of fire clay is used for this purpose, usually a mixture of 60 per cent of calcined

Plant	Max. Boiler Evaporation lb.per hr.	Pressure	Type Bottom	Walls	Length	Width	Height	Volume
Gorgas #2	450,000	475	R. Hopper	Fin tube	30	21.7	37.92	24,700
Huntley #2	560,000	450	Slag tap	Bailey	20	35	32	22,900
Quindaro	180,000	400	Slag tap	Bailey	18	21.5	22	10,900
Kalamazoo	145,500	400	R. Hopper	Fin	16.3	21.7	24	8,700
Deepwater	330,000	1350	Slag tap	Bare tube	34.3	16.8	30	19,080
Trenton Channel	350,000	400	Hopper	Refractory A.C	34	23		25,140
Buck	325,000	-	-	Fin	29	25	31.5	20,560
Colfax	330,000	305	Hopper	Fin	20.5	27.8	25.5	14,800
Kneeland St. (Boston)	250,000	200	Hopper	Bailey	18.9	20.5	30.8	11,900
So. Meadow	450,000	450	Hopper	Bailey	30	30	33.8	19,800
So. Amboy	270,000	1400	Slag tap	Bailey	18	27	23.8	11,542
Atkinson	450,000	450	Hopper-W.Cool.	Fin	30	21.5	38.9	19,800
Lakeside	300,000	1300	-	Fin	23	35	36	30,100
Port Washington	690,000	1300	Hopper	Fin & R.H.Sup.	55	24	40	55,000
Somerset	140,000	400	W. So. Hopper	R - air cool	18.5	29	22	12,420
South Street	240,000	400	R. Hopper	Bailey & Refr.	24	18.4	32	11,970
East River	250,000	385	R. Hopper	Fin	22	24.2	24.8	15,880
East River	1,000,000	385	R. Hopper	Fin	24	43	66.8	38,200
Glenwood #2	275,000	405	Hopper	Bailey	21.8	18.8	28.5	11,700
Glenwood	300,000	405	Hopper	Bailey	21.8	18.8	28.5	11,700
Schenectady-Stm.	350,000	450	Slag tap	Bailey	26.2	23.3	23.9	13,700
Gilbert	250,000	1350	-	Fin	18	31.3	32.2	18,100
Waukegan	300,000	675	Slag tap	Bailey	23.3	20.2	30.6	12,840
Waukegan	495,000	675	Slag tap	Bailey	36.2	20.2	34.1	23,960
Michigan City	345,000	720	Slag tap	Bailey	19.8	26.2	30.6	14,130
Horseshoe Lake	130,000	300	Hopper	Bailey	17	17.8	17.3	5,270
Horseshoe Lake	250,000	300	Slag tap	Bailey	16.6	23.5	34	13,230
Horseshoe Lake	150,000	300	Hopper	Bare tube	19.2	20.4	21	8,000
Burlington	525,000	730	Hopper		22	32.9	33.7	24,360
Valmont	275,000	360	Slag tap	Bailey	23.8	20	25	10,510
Comal	250,000	425	Refractory	-	22.5	26	30	16,800
State Line	450,000	650	Hopper	Bailey	-	-	-	21,000
Powerton	275,000	650	Hopper	Bailey	-	-	-	12,800
Nashville	175,000	425	Hopper	Ref.-Fin	17	18	26.5	8,000
Trinidad	325,000	412	Hopper	Fin	25.1	24.8	32	19,500
Cahokia	220,000	345	Bailey bare	Bare tube Bailey Fin	16.7	23	22	13,275
Hell Gate	400,000	300		Fin	26.4	25.1	35	18,800
Hell Gate	800,000	300	Slag tap	Bailey	34	41	30	40,770
Riverside	140,000	450	Slag tap	Bailey Bl. & stud tubes	16.3	17.7	20.4	8,660
Hampton	80,000	200	Flat	Air cooled	22.3	12.6	17	4,870
Bremo	200,000	525	Slag tap	Wt. tubes	17	20	29.3	8,611
Edgewater	300,000	675	Slag tap	Bailey	20	20	23.3	10,269
Average	331,212				23.6	24	29.7	17,210

TABLE I

FURNACE DATA ON MODERN INSTALLATIONS

In this table is shown data on 43 of the more important stations constructed in recent years. Nearly all of these furnaces it will be noted are fitted with some type of water cooled wall. The average furnace volume is 17,210 cu. ft. and the average of furnace dimensions is a furnace about 24 ft. sq. and 30 ft. high.

TABLE II

FURNACE VOLUMES AND HEAT LIBERATION IN MODERN FURNACES

Here are shown furnace volumes in a number of modern installations. Note that the average for stokers is only about half that for pulverized coal and for gas and oil furnaces. The average heat liberation for these installations is about 30,000 B.t.u. per cu. ft.

Year	Plant	Method Firing	Boiler H.S. Sq. Ft.	Type Furn. Walls	Furnace Vol. Cubic ft.	Furnace Vol. Cu.Ft./ Sq.Ft.H.S.	Max. Heat liberated B.t.u. / cu.ft per hr.
1927	Harbor Pt.	Stokers	16,100	Water Walls	6,850	0.425	20,200
1927	Bayside	Pulv. Coal	13,010	Water Walls	10,700	0.825	20,000
1927	Neches	Oil & Gas	15,300	Water Walls	8,050	0.525	-
1927	East River	Pulv. Coal	14,809	Water Walls	20,857	1.405	19,000
1927	Des Moines	Stoker	15,018	Water Walls	6,183	0.410	-
1927	Lauderdale	Oil	17,440	Water Walls	15,600	0.895	-
1927	Lakeside	Pulv. Coal	28,532	Water Walls	30,100	1.085	10,500
1928	Hell Gate	Pulv. Coal	23,800	Water Walls	18,000	0.756	31,000
1928	Cahokia	Pulv. Coal	18,010	Water Walls	12,000	0.666	13,000-27,000
1928	Trinidad	Pulv. Coal	19,471	Water Walls	18,540	0.953	14,300
1928	Saginaw River	Stoker	12,340	Water Walls	6,000	0.485	-
1928	Long Beach	Gas & Oil	34,162	Water Walls	19,800	0.580	18,500
1928	Hell Gate	Stokers	23,764	Water Walls	8,500	0.358	31,000
1928	Edgar	Stokers	15,093	Water Walls	9,996	0.626	18,000
1929	Ohio River	Stokers	11,520	Water Walls	5,600	0.486	35,000
1929	Fremont	Stokers	6,080		1,600	0.263	27,500-37,500
1929	East R. Add.	Pulv. Coal	60,014	Water Walls	38,200	0.635	29,400
1929	Delray #3	Stoker	23,978	Water Walls	13,800	0.577	33,000
1929	State Line	Pulv. Coal	10,294	Water Walls	22,500	2.180	-
1929	Station A	Gas & Oil	10,699	Water Walls	15,100	1.410	43,300
1929	Holland	Pulv. Coal	7,928	Water Walls	16,500	2.080	20,000
1930	Belle Isle	Pulv. Coal	12,670	Water Walls	6,700	0.530	31,000
1930	Louisiana	Oil & Gas	13,817	Water Walls	16,800	1.210	23,000
1930	So. Amboy	Pulv. Coal	7,214	Water Walls	12,900	1.790	26,500
1930	Deepwater	Oil & Gas	7,778	Water Walls	14,000	1.800	33,800
1930	Concepcion Road	Oil & Gas	4,990	Water Walls	7,018	0.141	39,474
1931	Raritan River	Stokers	11,810	Water Walls	6,500	0.550	39,400
1931	Hudson Ave.	Stokers	24,450	Water Walls	14,000	0.572	54,770
1931	Glenwood	Pulv. Coal	20,036	Water Walls	11,700	0.584	43,720
1931	Mich. City	Pulv. Coal	7,197	Water Walls	14,700	2.045	29,350
1931	Lakeside	Pulv. Coal	23,640	Water Walls	28,573	1.210	14,100
1932	Kidder	Stokers	7,500		2,200	0.293	64,000
1932	Pasadena	Gas & Oil	15,186	Water Walls	7,600	0.500	36,700
1932	Quindaro	Pulv. Coal	16,267	Water Walls	10,900	0.670	23,800
1932	Dodge City	Gas & Oil	6,000		5,545	0.925	16,900
1932	Harding St.	Stoker	11,810	Water Walls	6,700	0.565	37,300
						Av. 0.858	Av. 29,713
					Av. for Stokers - 0.459 " " Pulv.Coal - 1.160 " " Gas & Oil - 0.886		

clay and 40 per cent of raw clay, both being of the same composition as the material used in the brick itself. Enough water should be used to give the mixture the consistency of a thick cream. The brick should be dipped in this then rubbed and tapped in place.

Fireclay is the most common material used in making refractory brick for power plant furnaces but

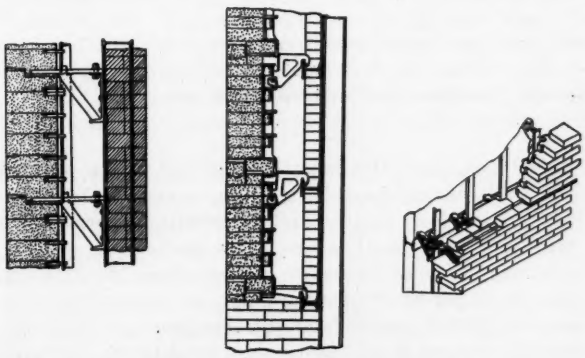


FIG. 4. TYPES OF SECTIONALLY SUPPORTED WALL CONSTRUCTIONS

silicon carbide is also used. This latter material has distinctive properties which make it superior to other materials for severe operating conditions. Insulating materials used in furnace walls are of two kinds, first, refractory gases and second, those which must be used behind separate or more refractory inner linings. There are a number of insulating materials available most of which, however, can be grouped under four different classifications. These are (a) asbestos fiber; (b) diatomaceous earths, (c) alumina and (d) kaolin.

Air Cooled Walls

Air cooled refractory walls may be classified according to two different types of construction, those which are self supporting or they may be of the sectionally supported type in which special refractory blocks are supported from structural steel members. Types of the first case are shown in Fig. 3, while in Fig. 4 several forms of sectionally supported wall constructions are shown. In most cases, where self supporting construction is used some form of special hollow or cast block as shown in Fig. 3 is used but an air cooled construction using standard brick is also possible, using two walls an inner one of fire brick and outer of common brick with suitable bonding between. The anchorage used between the walls in such construction is usually of a kind to permit of free expansion of one wall with respect to the other. An example of this type of construction is shown in Fig. 5.

In most cases, regardless of the type of construction used, those portions of the furnace walls adjacent

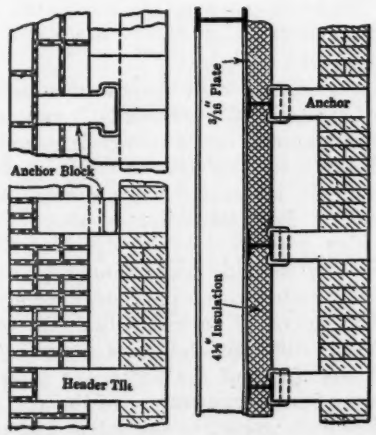


FIG. 5. SELF SUPPORTING AIR COOLED REFRACTORY WALLS USING COMMON BRICK AND FIRE BRICK

TABLE III. RECOMMENDED SETTING HEIGHTS FOR VARIOUS TYPES OF BOILERS EQUIPPED WITH STOKERS

Type of Boiler	Measurement of Setting Height
Water Tube	
Horizontal	Floor line to bottom of header above stoker
Inclined (Horizontal Mud Drum)	Floor line to center of mud drum
" (Vertical Mud Drum)	Floor line to top of mud drum
Vertical (Horizontal Mud Drum)	Floor line to center of mud drum
" (Vertical Mud Drum)	Floor line to top of mud drum
Horizontal Return Tubular	Floor line to under side of shell.

Abbreviations:
H.M.D. = Horizontal Mud Drum
V.M.D. = Vertical Mud Drum
H.R.T. = Horizontal Return Tubular
Min. = Absolute Minimum
Pref. Min. = Preferred Minimum, i.e., the minimum heights recommended

		TYPE OF STOKER											
		UNDERFEED				CHAIN GRATE				OVERFEED			
		Multiple Retort		Single Retort		Natural Draft		Forced Draft*		Side Feed		Front Feed	
		Min.	Pref. Min.	Min.	Pref. Min.	Min.	Pref. Min.	Min.	Pref. Min.	Min.	Pref. Min.	Min.	Pref. Min.
WATER TUBE	Horizontal, all sizes	11'-0"	13'-0"	9'-0"	11'-0"	10'-0"	12'-0"	12'-0"	14'-0"	9'-0"	11'-0"	9'-0"	11'-0"
	Inclined (H.M.D.) all sizes	7'-6"	8'-6"	6'-6"	8'-6"	6'-0"	8'-0"	7'-0"	8'-0"	5'-0"	7'-0"	6'-6"	8'-0"
	Inclined (V.M.D.) all sizes	6'-0"	7'-0"	5'-0"	7'-0"	4'-0"	5'-0"	6'-0"	8'-0"	3'-6"	5'-0"	4'-0"	5'-6"
	Vertical (H.M.D.) all sizes	3'-6"	5'-0"	3'-6"	5'-0"	3'-6"	4'-6"	4'-0"	5'-0"	3'-0"	4'-0"	3'-6"	5'-0"
	Vertical (V.M.D.) 150 hp.	4'-6"	5'-0"	4'-6"	5'-0"	4'-6"	5'-0"	5'-0"	5'-6"	3'-3"	.....	3'-6"	4'-6"
	Vertical (V.M.D.) 250 hp.	5'-6"	6'-0"	5'-6"	6'-0"	4'-6"	5'-0"	5'-0"	5'-6"	3'-3"	.....	3'-6"	4'-6"
	Vertical (V.M.D.) 500 hp.	6'-0"	6'-6"	6'-0"	6'-6"	4'-6"	5'-0"	6'-0"	6'-6"	3'-3"	.....	3'-6"	4'-6"
H.R.T.	72"	8'-0"	10'-0"	7'-6"	8'-6"	7'-0"	8'-0"	8'-0"	10'-0"	7'-0"	8'-0"	6'-0"	8'-0"
	84"	8'-0"	10'-0"	7'-6"	8'-6"	7'-0"	8'-0"	8'-0"	10'-0"	7'-0"	9'-0"	6'-0"	8'-0"

* When burning coke breeze and anthracite fines, the setting heights indicated above should be materially increased to provide for proper arch and furnace design.

to the fuel bed require particular attention and usually some special form of construction is used. At these points the walls are subjected not only to the heat of the furnace but also to the mechanical action of projecting clinker adhesions. For this reason special slag-resisting blocks of various kinds are used at these points. Sometimes these blocks are solid but more often they are air cooled.

Without going into the details of construction of all types of air cooled furnace walls, it may be assumed that if the furnace is to be stoker fired and designed for a moderate heat liberation of not more than 25,000 B.t.u. per cu. ft. air cooled walls should prove satisfactory. If the heat liberation is above that, water cooled bridge walls should be considered. The air used in cooling the side walls should be returned to the furnace for combustion purposes. With pulverized coal, natural gas or oil or with kiln-dried wood refuse the problem is different and more complicated. Here the length and form of the flame, the load factor of the boiler, the fusion temperature of the ash, the design of the boiler with respect to the extent and position of radiant heating surface exposed must be taken into consideration and carefully studied. If the load factor is low and the boiler must be operated at low ratings the walls should have little if any water cooling surface. High load factors and high rating operation on the other hand justify more water cooling. In general, for rates of heat liberation of less than 15,000 B.t.u. per cu. ft. per hr. the furnaces may be constructed with air cooled side walls and ash hoppers provided the furnace bottom is sufficiently depressed below the flame travel to prevent slagging of the ash. These questions are discussed in detail by C. S. Gladden in a paper presented before the A.S.M.E. in 1929[2]. If the heat liberation is between 15,000 and 20,000 B.t.u. per cu. ft. per hr. a water screen in the furnace bottom and a cer-

[2]Proprietary air cooled refractory walls. Transactions A.S.M.E. FSP-51-40.

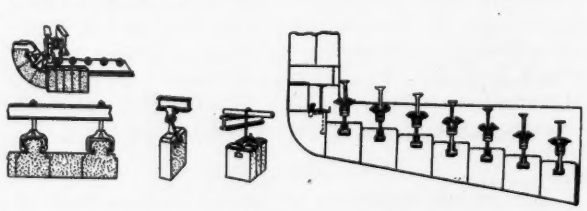


FIG. 6. FLAT SUSPENDED ARCHES

TABLE IV. MAXIMUM HEAT RELEASE VALUES FOR PULVERIZED COAL FURNACES

Ash Fusing Temperature	Below 2100 Degrees Fahr.	2100–2400 Degrees Fahr.	Above 2400 Degrees Fahr.
Solid refractory walls	Never	12,500	15,000
Air-cooled refractory walls	12,000	15,000	17,500
Water-cooled bottom	16,000	17,500	20,000
Water walls	17,500	22,000	30,000

tain amount of water cooling of the side walls is advisable.

From what has been pointed out here it is evident that the selection of the type of furnace for a particular installation is not a matter of personal like or dislike for one type or another, but should be based on a thorough engineering analysis of conditions. The air cooled wall has not displaced the solid refractory wall, nor has the water wall displaced the air cooled wall. While the lines of demarcation between the applications of these various types is not sharp, there is a definite field for each. In some instances combinations of two types of construction may be used to advantage.

Arches

No discussion of furnace design or construction would be complete without a consideration of arches. Various forms of arches are used in furnaces, particularly those of the stoker fired type, to aid the combustion. Their specific function is to reflect radiant heat from hot parts of the fuel bed to such portions where the gases tend to become chilled below the combustion temperature. The use of arches is particularly desirable in the case of chain grate and overfeed stokers—those in which fresh coal is exposed to the full furnace temperature. With underfeed stokers, where the coal is introduced at the bottom and the air also there is a thorough mixing of air and volatile gases and a gradual heating as the mixture rises to the surface so that usually no arches are necessary with this type of stoker.

Modern arch construction follows the same fundamental principles used in refractory furnace walls. They usually are of the air cooled type and supported independently from steel members in the furnace. Originally, arches were of the curved or sprung type, supported from the side walls by virtue of the arch principle in masonry, hence the name. This type of arch requires great care in construction and when repairs are necessary the unit must be shut down for considerable periods. For these reasons, the flat, suspended, air cooled type of arch has been developed. Several forms of these arches are shown in Fig. 6. They are flexibly supported, have adequate provision for expansion and contraction, and carry no load except their own weight. Another advantage inherent in their use is the fact that individual blocks or small groups can be easily and quickly replaced without disturbing other blocks.

Refractory materials used in each construction are the same as those used for furnace wall construction, fire clay brick, silicon carbide, etc. An arch of fireclay blocks provided with a surface veneer of silicon carbide is used to advantage in furnaces subject to extremely high temperatures and rapid temperature changes. It is quite effective in withstanding local erosion.

The shape and location of arches naturally is dependent upon many factors, involving the entire design of the furnace. As stated previously, arches are not generally used with underfeed stokers where proper settings are provided. With chain grate stokers, the application of arches is of considerable importance as it has been found that the efficient burning of different kinds of fuels depends on the length of the ignition and combustion arch, its height above and angle of inclination with the fuel bed. Ignition arches for chain grates vary from 3 ft. to 6 ft. in length.

Lighting Pulverized Fuel Furnaces

In the operation of pulverized fuel furnaces extreme care is essential at the time the fuel is lighted as disastrous explosions have taken place during this operation and when for some reason the fuel supply was interrupted momentarily while the furnace was in operation. To forestall such accidents the N.E.L.A. has recommended a code giving instructions for doing this lighting which is as follows:

Feeders—Starting

1. Make general inspection.
2. Start lubrication.
3. Start primary air fans.
4. Adjust primary air to operating pressure.
5. Open the dampers (gates) above the feeder.
6. Insert lighted torch in the burner.
7. Close the main feeder motor switch.
8. Open the primary air damper on the feeder.
9. Start the feeder motor with the clutch out.
10. Throw in the feeder-motor clutch.

Feeders—Stopping

1. Close gate over feeder screw.
2. Stop feeder, after running the hopper empty.
3. Stop lubrication.
4. Close primary air damper.
5. Shut down primary air fan.
6. Open the main feeder switch.

Mills and Burners (Unit System)—Starting

1. Furnace operator should make a thorough inspection of mill.
2. Check for proper operation of magnetic separator.
3. Start lubrication.
4. Start mill with coal supply closed.
5. Regulate boiler damper or induced draft fan to give an under pressure in combustion chamber.
6. Light and insert ignition torch through port provided.
7. Start coal feed to mill at a rate determined by experience for best ignition.
8. Remove ignition torch as soon as burners are properly lighted.
9. Increase feeder speed as required.

Mills and Burners (Unit System)—Shutting Down

1. Stop coal feed to mill.
2. Allow all coal to run out of mill.
3. Stop mill.
4. Stop lubrication.

POWER PLANT ENGINEERING January 1934

High Ratings *made Possible by use of* WATER WALLS

The development and use of the water cooled furnace has enormously increased the ratings at which boilers may be operated. The water wall is not the solution to all furnace problems however and a careful study of operating conditions should govern its selection.

IN THE PRECEDING article, we considered briefly the two types of refractory wall construction used in furnaces and some of the factors involved in their selection. It was shown that for furnaces of low or moderate capacity the solid refractory wall was the most economical. For furnaces of higher capacity the air cooled wall was more suitable. Now we shall consider the third type of furnace wall construction, the water wall type, suitable for high capacity, high temperature units where prolonged intensive rates of firing prevail.

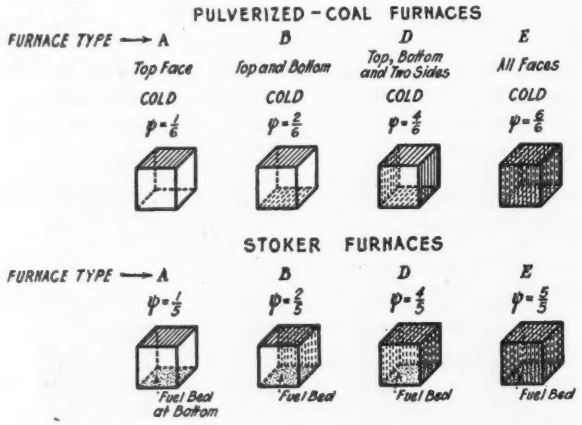


FIG. 1. CONVENTIONAL REPRESENTATIONS OF FURNACE TYPES

As was implied in the preceding article, the combustion rate of fuel in a furnace is limited only by the speed with which the elements involved will combine. This limiting rate is so high and present rates so low, relatively speaking, that most research in furnace design in recent years has been towards increasing the speed of combustion without undue wear on the furnace walls. The water cooled wall is an effort in this direction, and in the years that they have been in use they have proved singularly successful.

Water cooled walls, while they are not subject to the inherent limitations of refractory walls are more expensive and normally are used only in instances where refractory surfaces would give trouble. In many cases only certain portions of a furnace need be water cooled. In a furnace, for example, fired with a multiple retort underfeed stoker and operated at moderately high ratings only the bridge wall or back wall needs to be water cooled for this receives the greatest punishment. With chain grate stokers as a rule only the walls along the line of the fuel bed require protection, but when operated at very high rating all the walls and arches may require water cooling. So also, furnaces fired with powdered coal, oil or gas may be partly or entirely water cooled depending upon conditions. The question of water cooled walls in other words is not one of method of firing or type of fuel used, but rather one of heat release and furnace temperature.

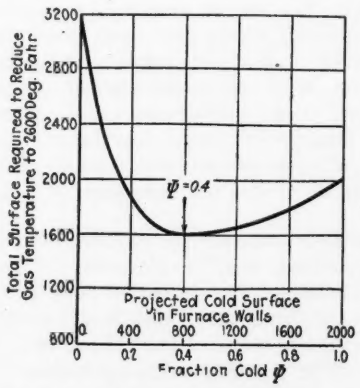


FIG. 2. INFLUENCE OF "FRACTION COLD" ON TOTAL SURFACE REQUIREMENTS

Furnace 8000 cu. ft., 80,000 B.t.u. per cu. ft. per hr. Illinois coal; 4% excess air; stoker fired.

The fundamental elements of heat release and heat absorption have been studied quite thoroughly by a number of investigators. Wohlenburg and Brooks, in a paper before the A. S. M. E.[1] showed that the totally cold, or water cooled furnace provides the highest permissible energy release rate for flames of given temperature. Their analysis of the various possible conditions is rather interesting and while we cannot present details in an article of this length, it may be of interest to show their classification of furnace types. In Fig. 1, the shaded faces of the conventionally shown cubical furnace cavities represent water wall surfaces

[1] A.S.M.E. Transactions FSP-50-39.

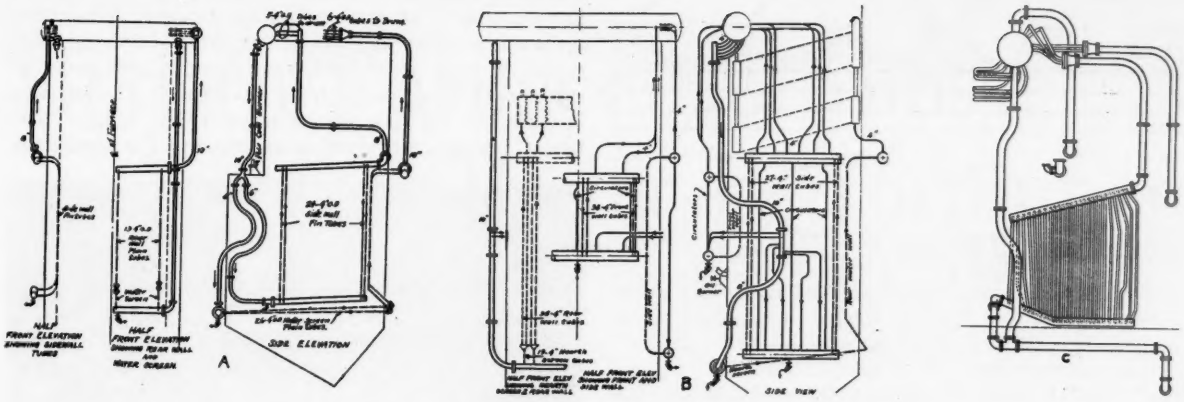


FIG. 3. WATER WALL CONNECTIONS

(a) In this arrangement a 10 in. downcomer is connected to the boiler by six 4 in. tubes. (b) Here five 4 in. tubes with long bands to take care of expansion, supply an 8 in. downcomer from each end of the boiler drum in this arrangement. (c) In this system a 10 in. downcomer is taken directly off each end of the boiler drum.

and the remaining faces in each case refractory surfaces. The type A pulverized coal furnace has one face cold or water walled and five refractory faces. To define this condition, the term "fraction cold", has been introduced. In this case, the fraction cold is equal to one-sixth. In other words it is the fractional part of the furnace envelope which is occupied by water-wall surfaces. The type B furnace has two water-wall surfaces, hence a fraction cold of two-sixths. The type E or all-cold-walled furnace has six cold faces for pulverized coal and five cold faces for the stoker, hence the difference in the two cases for the types A, B and D furnaces.

Curve Fig. 2 shows how for the conditions indicated, the total surface requirement for a given final temperature varies with the fraction cold. The least requirement results when it is about four-tenths or when 800 sq. ft. of surface is placed in the furnace envelope. This arrangement would correspond closely to that shown for the type B furnace in Fig. 1. Under these conditions the total surface requirement for a 2600 deg. F. gas temperature is 1630 sq. ft., whence the difference 1630 — 800 = 830 sq. ft., is placed in the convection zone. If the fraction cold is either larger or smaller than that required for the minimum point on the curve it will obviously require more total heating surface, but it is seen that for all arrangements resulting in values of fraction cold between two-tenths and unity, the variation is not large.

Water walls used today are of various types. For convenience they may be classified under three types, however, 1 bare plate walls, 2 bare tube walls and 3 covered tube walls. As a rule, however, in speaking of water walls, the bare plate type is not considered, for it was developed not primarily as a water wall but as a part of certain types of boiler design. The cylindrical furnaces in fire tube boilers of the Scotch marine type and in the furnaces in locomotive and small vertical type boilers are all water cooled, but they are a part of the boiler itself.

By the term water wall as used here we mean the development of placing water backed surfaces in the furnace exposed to direct radiation from the incandescent gases. These surfaces may be made up of bare tubes or of tubes covered with metallic or refractory blocks. These tubes are usually connected into some part of the boiler proper, so that the heat absorbed by the water in the tubes is finally delivered to the boiler. In some cases the walls have been made with independent systems of their own. A number of different methods of connection are in use, several methods being shown in Fig. 3. As a rule the heating surface of the walls are connected into a lower header which is supplied with cool water from an overhead drum by means of one or more downcomer tubes and the heated, less dense water discharged from the tubes, together with any steam entrained in it, is delivered from the upper header to the same drum by means of risers.

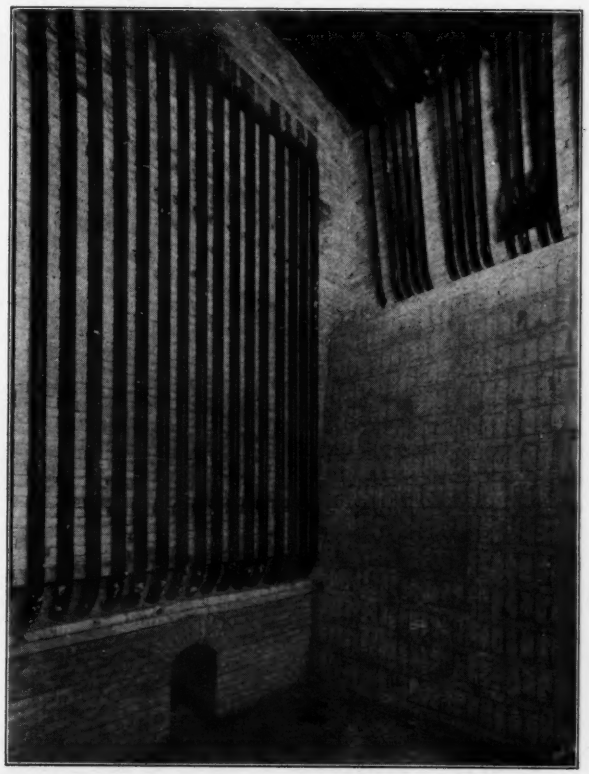


FIG. 4. SIMPLE TYPE OF WATER WALL

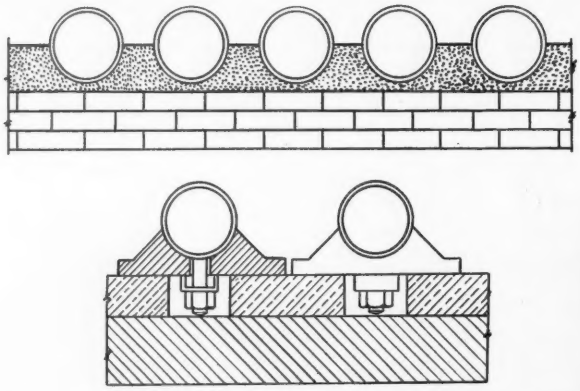


FIG. 5. ARRANGEMENT OF BARE TUBES IN WATER WALLS

In Fig. 4 is shown a simple type of bare water wall which is economical in construction and effective. As can be seen there are simple straight tubes spaced a small distance apart and located in front of an ordinary refractory wall which they partly screen and from which the rear of the tubes receive radiated heat. These tubes are entirely on the surface of the refractory. In other forms of bare tube construction the tubes are imbedded partially, usually to the extent of about half their diameter in refractory thus permitting transfer of heat from the refractory by conduction direct to the tubes. Another form is the so-called wing back construction shown in Fig. 5B, in which machined castings are attached to the backs of the tubes by means of studs welded to the tubes. This not only increases the heat transmission to the tubes but provides increased protection to the refractory walls.

In the example shown the tubes are spaced some distance apart. In other types the tubes are arranged so that there is practically an unbroken area of water

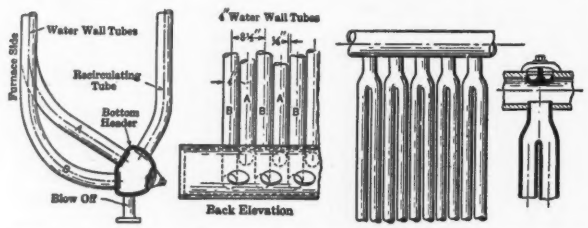


FIG. 6. METHODS OF OBTAINING WALLS WITH CLOSELY SPACED TUBES

tube surface exposed to the furnaces. Such an arrangement is shown in Fig. 6 and is accomplished by connecting the tubes into the headers at different points, or by using special bifurcated tubes. Still another scheme for obtaining practically continuous water cooled surfaces is the use of fin tube construction. A furnace using this construction is shown in Fig. 7 and the details of construction are shown in Fig. 8. In this type two longitudinal fins are electrically welded to opposite sides of the tube. These tubes are then placed so that the fins are all in one plane, and the whole backed by the usual refractory wall. If desired a thin backing of heat insulating material can be used instead of the thicker refractory as shown in Fig. 8.

The advantage of covered walls is that it protects the tubes from damage due to blistering and permits operation at lower rates and over a wider range than would be possible with bare walls. It is true, in furnaces operated at extremely high capacity bare tubes are used to advantage, but at low rates of driving, particularly with pulverized coal firing, they may introduce serious losses from incomplete combustion, because the ignition period is retarded. Higher preheated air and turbulent firing improve the conditions

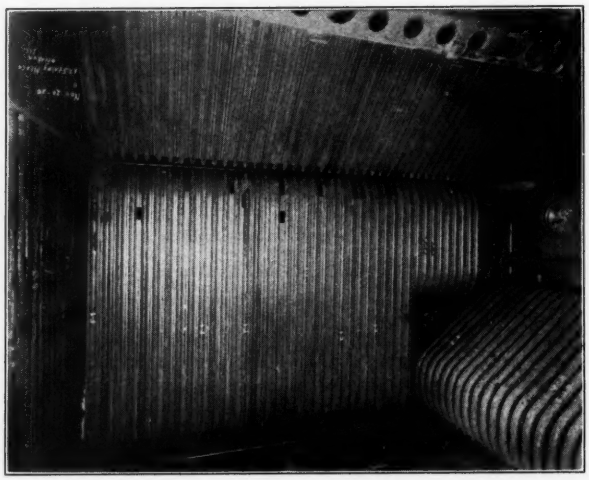


FIG. 7. FIN TUBE WATER WALL

in this respect at the higher rates of driving the furnace. In most instances, the design of the furnace must be a compromise between the conditions which provide for good combustion at all rates of driving and those which provide for lowest cost of maintenance under the conditions of operation that must be met. This quite often resolves itself into a structure in which there are some bare iron water cooled walls and others of the uncooled refractory type. In another case a part or all of the inner faces of such walls may be of refractory backed by water cooled surfaces. Only for large furnaces, for small furnaces to be operated continuously at very high ratings, or in cases where extra fine grinding of coal in pulverized fuel firing is to be used, should consideration be given to furnace design involving entirely bare iron water cooled surfaces.

Water cooled furnace walls do not appreciably increase overall efficiency through heat absorption and reduction in temperature of furnace gases but rather through making it possible to carry higher percentages of CO_2 than would otherwise be possible. The expense of water cooled furnace walls is justified only through savings in coal and cost of equipment. Cost of maintenance is reduced. Many companies reporting on operation, indicate a reduction of furnace and stoker maintenance costs due to the installation of water cooled walls. In one particular instance a reduction of maintenance costs to the extent of 50 per cent is reported.

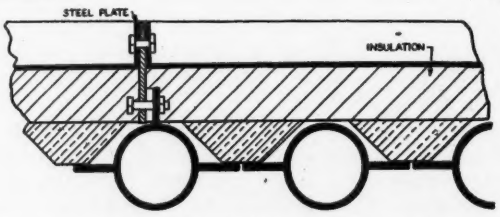


FIG. 8. DETAILS OF FIN TUBE CONSTRUCTION

Modern Furnace Bottoms *improve* Boiler Performance

FEW things in Power Plant Design have undergone such a revolutionary change as have our ideas in regard to furnace bottoms, particularly those for high capacity pulverized coal furnaces. All our previous efforts in regard to furnace bottom design were toward preventing fusion of the ash. In the modern slag tap furnace, conditions are set up to deliberately cause fusion of the ash and thus permit its removal from the furnace as a liquid.

LOGICALLY, there is no reason for considering the design of the furnace bottom apart from the design of the furnace proper, for the bottom is just as much a part of the furnace as are the other surfaces of the furnace envelope, and its design is tied up integrally with the design of the furnace itself. Its design will vary widely, depending upon the kind of fuel used, the characteristics of the ash, the methods of firing and the degree of forcing and several other factors. Everyone is familiar with the simple ash pit of the hand fired boiler and in the case of the small stoker, either of the chain grate or under or overfeed variety, only little more attention need be given the ashpit. Large and moderate size stokers, however, demand more careful study and here the ash is discharged into storage hoppers, lined either with refractory or with air or water cooled surfaces. Typical forms of ash pits used with stokers are shown in Fig. 1. As shown, generally the ash deposited in these hoppers is quenched by using a water spray as it is removed. With stokers and for use with certain coals clinker grinder rolls are placed at the entrance to the ash hoppers which reduce the size of the ash delivered to the hoppers to a uniform size. In other cases dump grates normally cover the hopper entrance so that all combustible will be thoroughly consumed before the ash is delivered to the hopper.

With pulverized coal, the design of furnace bottoms is more important because in this case improper design is likely to be very troublesome.

For in pulverized coal furnaces conditions are quite different from those existing in stoker fired furnaces. When the fusing temperature of the ash in the coal is higher than the furnace temperature, the ash remains as dust. This is deposited on the furnace bottom as well as on the walls and heating surfaces. Where the fusing temperature of the ash is appreciably below the furnace temperature the particles of ash become molten globules which adhere tenaciously to almost any part of the furnace or heating surface with which they come in contact. That which accumulates on the furnace walls, melts and flows downward, fluxing and eroding any hot refractory material that may be present. It is this second condition which makes the design of the powdered coal furnace bottom so much more important for the accumulation of this ash on the furnace bottom is likely to build up into a solidified mass very difficult of removal. These molten particles will also adhere to the boiler heating surfaces, building up slag and spongy masses which often choke the gas passages.

In modern practice, the removal of coal from pulverized coal furnaces is accomplished in four different ways[1]:

1. The admission of cooling air near the bottom of the furnace so that the ash falling into this zone is chilled before conglomerating into a solid mass on the floor proper. Such floors may consist of flat refractory bottoms from which the ash is hoed or sluiced out. This method has its limitations as to furnace temperature and ash fusion.

2. A similar furnace bottom, either flat or hopper bottomed with water cooling tubes (so called water

[1]See, paper the slag tap furnace and its effects upon the selection of coal for burning in pulverized form, by Bailey and Hardgrove, presented before the Third International Conference on Bituminous Coals.

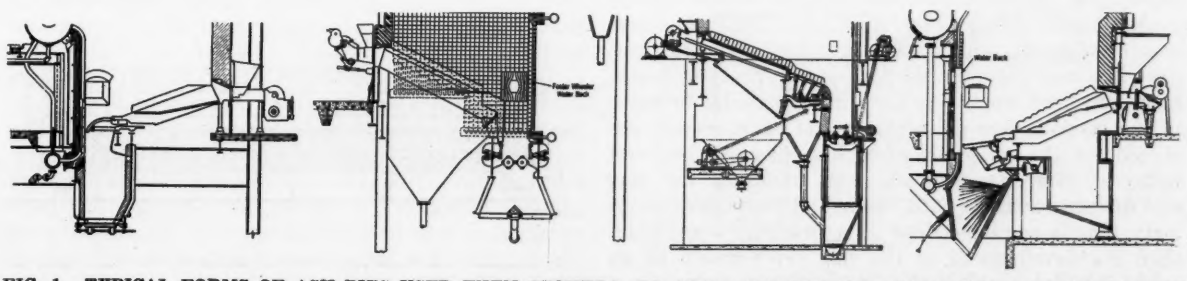


FIG. 1. TYPICAL FORMS OF ASH PITS USED WITH STOKERS TO MEET DIFFERENT LOAD AND ASH CONDITIONS

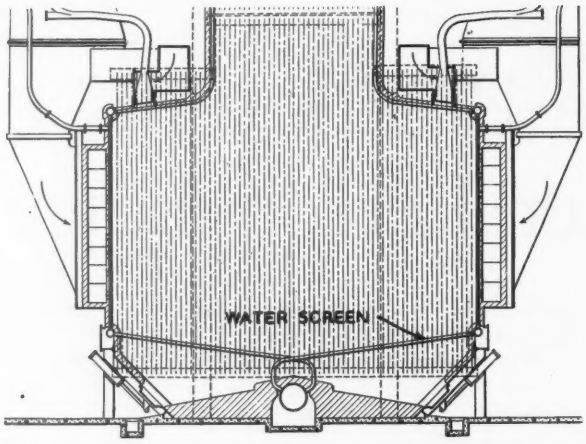


FIG. 2. DIAGRAM SHOWING POSITION OF WATER SCREEN

screen) spread across the furnace a short distance above the floor, so as to add to the cooling of the ash as it accumulates on the floor. This water cooling may be used to supplement the air cooling or in place of it.

3. A water cooled hopper bottom furnace, where the ash is chilled as it accumulates and where the floor is composed of relatively smooth cast-iron blocks or tubes, so that the ash falls or rolls by gravity into the hopper, without adhering to or accumulating into objectionable masses on the floor of the hopper. When low-fusion-ash coal is burned at high rates of liberation and high furnace temperatures, the ash usually flows down such a hopper bottom in a molten or semi-plastic condition and drips into the ash pit. This type of furnace bottom is more nearly universally applicable to the burning of both high and low-fusing-ash coal at high and low ratings than any other bottom so far developed.

4. A slag tap furnace wherein the ash which accumulates on the bottom can be retained in molten condition until periodically tapped out in liquid form. Such a furnace bottom naturally has no cooling by air or water tubes above the zone where the ash accumulates. On the contrary the flame from the burners can be directed immediately above the floor or be caused to impinge against the slag accumulating in this zone to keep the ash in molten condition.

It is this last type of furnace bottom that is perhaps the most interesting since it is an example of how a development which was tending in one direction, can often be solved by proceeding in a directly opposite direction. In the 1926 extension to the Charles Huntley Station at Buffalo, it was noted that pools of slag accumulated in a molten condition at the bottom of the furnace well, despite all attempts to keep the well walls as cool as possible. This, of course, was undesirable, but it presented the idea that perhaps this was the very solution of the problem everybody had been working on. So the remaining furnaces in the station were deliberately and courageously designed for the collection of molten slag on their wet bottoms, after experiments with handling the slag and disintegrating it with water had convinced everybody that it could be done in a practical way. Since then the development of the slag tap furnace as we know it today was rapid.

Before we discuss the details of the slag tap furnace a brief consideration of the second method, that using the plain water screen will be of interest. The water screen in the modern furnace consists of a row of water tubes spaced far enough apart, usually around 14 in., so that ash or slag can readily drop between them. In falling through this screen the particles are cooled and solidification occurs before the ash reaches the bottom of the furnace. This arrangement has been quite successful, though at heavy loads the flame may penetrate the screen and slagging on the tubes and in the ashpit may result.

Several diagrams showing the construction of slag tap furnace bottoms are shown in Fig. 3. Generally they consist merely of flat or gently sloping floors of several layers of refractory material and insulating material and provided with a tap at one side for drawing off the liquid ash. In some cases the floor is water cooled.

The slag tap furnace bottom can be used successfully with all coals in which the ash has a fusing

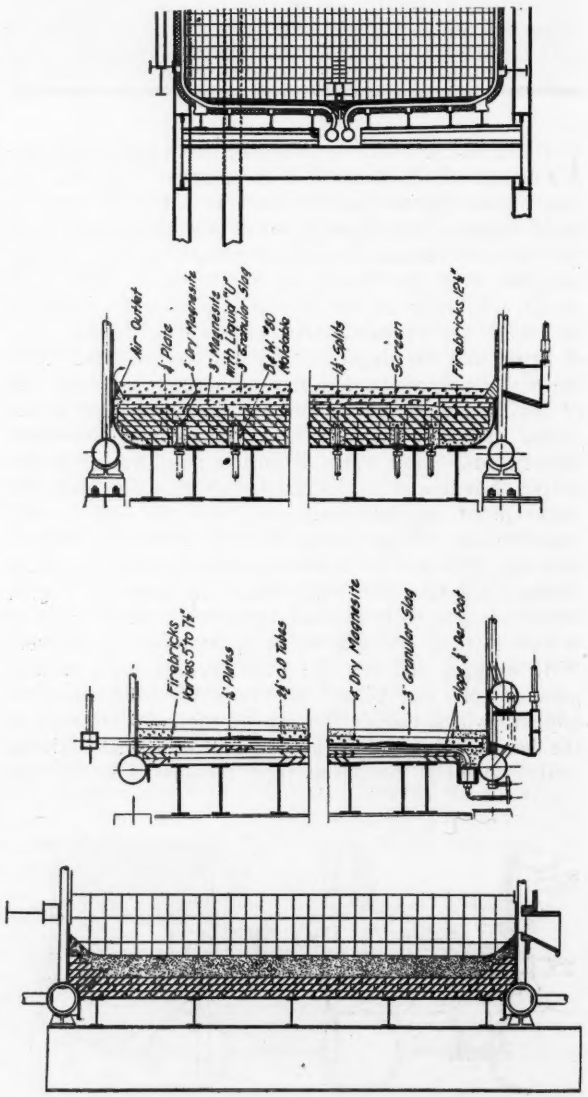


FIG. 3. SLAG TAP FURNACE BOTTOMS

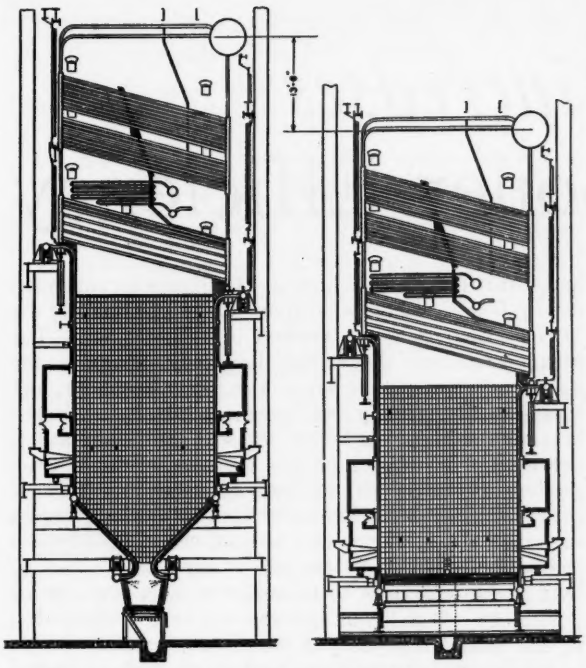


FIG. 4. SHOWING HOW THE USE OF THE SLAG TAP BOTTOMS EFFECT A REDUCTION IN THE OVERALL HEIGHT OF AN INSTALLATION

temperature below 2500 deg. F. and most of the coal now used falls in that range. At the Huntly station coal of about 12 to 14 per cent ash with a fusing point of around 2000 deg. was burned successfully with heat release rates exceeding 40,000 B.t.u. per hr. per cu. ft. of furnace and with only 18 per cent excess air. Another advantage, is its ability to burn coal completely. Coarse coal can be fed without increasing the combustible loss, as these coarse particles float on the surface of the molten slag until they are consumed. In the case of the furnace shown in Fig. 3 the flame from the burner, pointed downward, impinges directly on the molten slag which provides an ideal target. The only precaution to be taken in the application of this principle is that the burners must be far enough away from the surface of the slag that the tarry matter will be distilled from the coal and only dry particles of coke will impinge against the slag.

Saves Building Height

Another feature of the slag tap furnace that presents advantage is that it permits a reduction in the height of the furnace with a consequent saving in building costs. This is shown in Fig. 4. Here are two boilers of identical construction with furnaces of the same width, depth and method of firing, one with a slag tap bottom furnace and the other with a hopper bottom furnace. In this comparison there is an actual saving of 13½ ft. in the building height.

Early liquid bottom furnaces were constructed with a flat steel plate bottom covered with one or more layers of firebrick on top of which 6 in. or more of burned dolomite was added. Later sand bottoms were tried and still later the air and water cooled bottoms were developed. A typical air cooled bottom is shown in Fig. 3B. A modern water cooled bottom is shown at 3C. The air cooled bottom consists of a ¼ in. flat steel plate with the sides curved up. Firebrick was laid 12½ in. thick on top of the plate and the brick in turn was covered with 3 in. of magnesite mixed with a special liquid. A layer of 1¼ in. split brick covers this mixture with a slope toward the slag tap hole. A mesh screen is placed on the split brick and 3 in. of granular slag added for original joining. At frequent intervals throughout the bottom vertical perforated pipe nipples are installed, extending some 6 in. above the bottom plate. These pipes are connected with headers to a motor driven compressor developing about 29 in. of air pressure to insure an even distribution to all nozzles. The air is intended to flow from the nozzles to the split-brick "roof" and then to the side walls and through orifices into the furnace.

The water cooled bottom has the usual flat bottom plate above which is placed a layer of loose laid firebrick which varies from 5 to 7½ in. in thickness. The 2½ in. cooling tubes are laid in, extending from an outside header at this point of the boiler to another header at the top or downtake wall. These headers are connected to the boiler circulating system. Above the tubes are laid steel plates, overlapped to provide for expansion. Over the plates are 4 in. of dry magnesite and then 3 in. of granulated slag, all sloped toward the tap hole.

For handling the slag as it is removed from the furnace the principle shown in Fig. 5 has proved successful. As the slag emerges from the tap it falls on a water table formed by four primary jets of water where it is cut into small pieces and partly chilled. Falling from here into a U shaped cast-iron trough it is carried crosswise with the aid of two carrier jets for a distance of about 10 ft. where it hits a smashing plate and drops 6 ft. into another U shaped trough. This carries it away and finally discharges it into a sump. The slag is deposited in a fine granular state at the discharge of the chute.

Continuous Tap Method Just Developed

Recently a new type of furnace from which the slag is tapped continuously through a hole in the center of the floor was developed and is now being used in the Chicago district. Slag falling through the hole drops into a tank of water and is granulated, the tank being emptied every day. Cooling water makeup is added automatically. This form promises to further increase the popularity of the slag tap furnace.

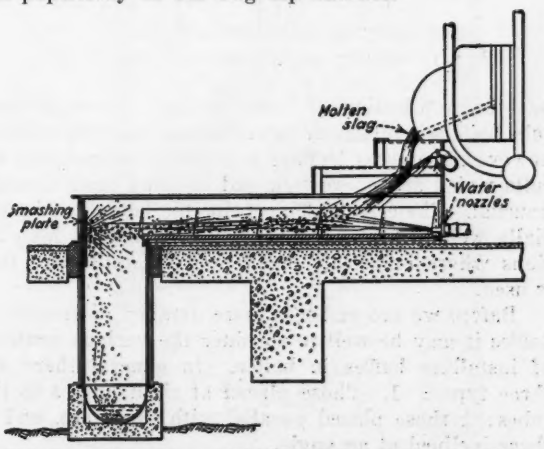


FIG. 5. A SLUICING SYSTEM USED WITH SLAG TAP FURNACE

POWER PLANT ENGINEERING January 1934

Proper Baffling *increases* Boiler Efficiency

UNTIL RECENT years, no feature in connection with the design and operation of water tube boilers received so little attention as that of baffling. And yet, the question of baffling is a highly important one since the design and placing of baffles have a marked influence upon the boiler efficiency. Indeed some engineers are of the opinion that the peculiarities of tube arrangement in boilers count for less than proper furnace design and proper baffling.

During the past few years, however, the question of baffling has received considerable attention and a great amount of study has been given it. These studies, for the most part, have been made by independent investigators and information on this subject is quite scattered and difficult of proper analysis. The several manufacturers of baffles, however, have done much to coördinate the knowledge regarding baffling and today the theory and practice of baffling is approaching some degree of uniformity. For a number of years the improvements that have been secured by the baffling of horizontal straight tube boilers has been understood,

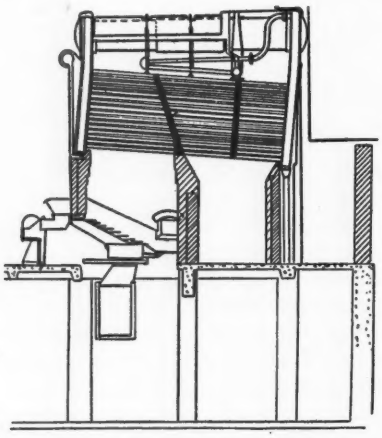


FIG. 1. BOILER WITH IMPROPER BAFFLING

but the application of cross baffles to vertical bent tube boilers is relatively new. With proper application, however, the cross baffling principle has produced results which are interesting and in some cases thought unusual. The cross baffling principle lends itself especially well to back arch construction and, in installations where extremely small sizes of anthracite fuel is used.

Before we proceed to a more detailed discussion of baffles it may be well to consider the various methods of installing baffles in boiler. In general there are three types: 1. Those placed at right angles to the tubes; 2, those placed parallel with the tubes, and 3, those inclined at an angle.

In the application of these general schemes, no hard and fast rules can be followed. The baffle arrangement in any boiler will depend upon the type of boiler, the type of fuel used, the method of burning, etc. The arrangement of the baffles largely determines the length of the gas passages through the boiler and therefore the velocity with which the gases sweep the heating surfaces. A boiler whose heating surface is arranged to provide long gas passages of small cross sections as a rule will be more efficient than a boiler in which the gas passages are short and of large cross section. In this way it will be seen the baffling affects the heat absorption by the boiler. It is also apparent that if the baffling is to be effective there must be no possibility of gas leakage through the baffles, that is, the gases must be compelled to travel the entire length of the gas passage rather than short circuit from one pass to another. Short circuiting of gases is a common fault and the baffling always should be inspected when the boiler is shut down for cleaning and repairs.

The absorption of radiant heat by the boiler to a large extent is controlled by the arrangement of baffling. Consider the boiler shown in Fig. 1. Compared to Fig. 2, which has a properly designed baffle, it can be seen that the volume of the furnace is too small and that the tube surface exposed to the radiant heat of the furnace is only about one half. These dia-

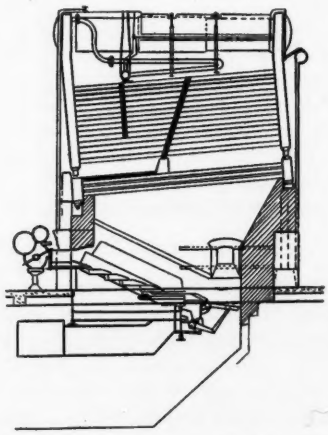


FIG. 2. SAME BOILER AS IN FIG. 1 BUT WITH REARRANGEMENT OF BAFFLES

grams represent the same boiler. Fig. 1 was revamped, raised, turned around, a new stoker installed and converted into Fig. 2. The normal rating was increased from 800 to 1250 hp. With a lower grade of coal than that used before, it was found that 13.2 per cent less coal was required per pound of steam.

In general, the underlying principles governing the location of baffles are as follows:

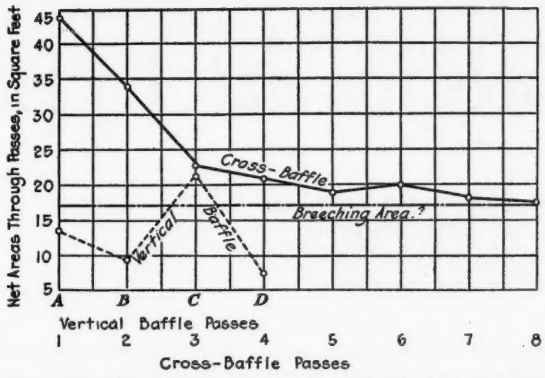


FIG. 3. CURVE SHOWING AREAS OF GAS PASSAGES WITH RESPECT TO BREECHING OPENING FOR TWO CONDITIONS OF BAFFLING

1. Tapering the gas pass to follow as closely as possible the reduced volume of the gases due to their shrinkage in cooling.

2. Securing a low entrance velocity which aids the draft and relieves clogging of the gas passage by the slag which may be present in the fuel.

3. Exposing the largest possible area of the first rows of tubes to the radiant heat of the furnace.

4. Placing as much heating surface in the first pass as conditions will permit.

5. Placing furnace surfaces in such position that the heat will be reflected out of the furnace to the heating surfaces instead of being held in the furnace.

6. Placing baffles so they will prevent the accumulation of soot.

7. Increasing combustion volume by moving back bridge walls, inclining them backward for rigidity or doing away with them entirely.

8. Relieving the resistance wherever the gas must be turned.

It is true that there are designs of boilers and furnaces where these ideas must be modified, but in the main they are correct.

These principles seem clear and logical enough, but it is only within recent years that they have been applied in practice. As with many other developments, the origin of cross baffling of vertical boilers was the result of trouble. It had been found that on a 400 hp. vertical boiler, rating could not be secured by any method of changing draft, increasing pressure on the burners, etc. Investigation finally indicated that the gases were not permitted to get away from the boiler. It was found from a study of the actual areas of various passes that the wet areas through the various points at which the gas was compelled to pass in getting from one bank of tubes to the next were from 10 to 50 per cent below the areas through the breeching. So by using this opening through the breeching as a preliminary starting point on which to base the various openings, a curve was prepared that would show this comparison of areas as plotted against the damper settings. This curve is shown in Fig. 3. It will be noticed that all but one of the openings in the original baffling were decidedly below the breeching area. The completed baffle as laid out in these curves is shown in Fig. 4. When this work was first done, the baffling design was based entirely upon the opening through the manufacturer's breeching.

The primary object in making this original installation was to remove the restrictions and allow an easier flow of gas through the boiler thus giving the gas burners an opportunity to function properly. In making the study, however, several other characteristics were noticed which promised to be of even greater importance than the primary object for which the study had been made. The principle of this was the arrangement of the heating surface with regard to the passes, as with the layout made in this manner, it was discovered that a materially greater amount of tube surface was made possible to be located in the first and second passes than was possible with the original method. It was found, too, that this percentage of heating surface varied with each particular type and in some cases with the different sizes of various boilers. This change in percentage of heating surfaces in one type of boiler with the original baffling was found to be as follows when tube surface was considered:

1st pass	35.8%
2nd pass	32.4%
3rd pass	31.8%

or very close to an equal division of heating surface in each of the three passes.

With the rearrangement of the baffle as shown in Fig. 3 the distribution of heating surface in the various passes was as follows:

1st pass	44.85%
2nd pass	29.4 %
3rd pass	25.75%

that is, an increase in the heating surface in the first two passes from approximately 68 to 75 per cent.

The complete analysis of the results obtained by these changes in this original installation are reported by A. C. Danks in A. S. M. E. transactions FSP—50-57, but they were satisfactory in every way and accomplished everything that was intended of them. It increased their rating, eliminated irregularities of the flow of gases, and "puffing" that is sometimes noticed with boilers operating on blast furnace gas.

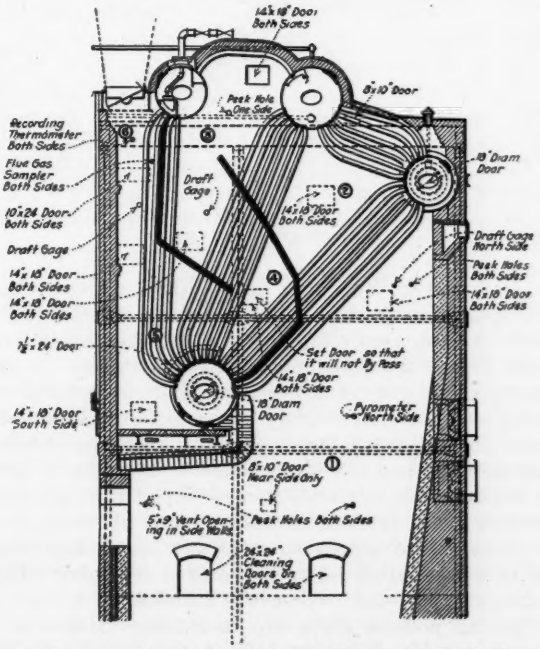

14½"x18" Door Both Sides
Peek Hole One Side
8½"x10" Door
Recording Thermometer Both Sides
Flue Gas Sampler Both Sides
Draft Gage
18" Diam Door
10"x24" Door Both Sides
14"x18" Door Both Sides
Draft Gage
Draft Gage North Side
14"x18" Door Both Sides
Peek Holes Both Sides
14"x18" Door Both Sides
14½"x10" Door Both Sides
1½"x24" Door
Set Door so that it will not By Pass
14"x18" Door Both Sides
14"x18" Door South Side
18" Diam Door
Pyrometer North Side
8½"x10" Door Near Side Only
5"x9" Vent Opening in Side Walls
Peek Holes Both Sides
24"x24" Cleaning Doors on Both Sides

FIG 4. CROSS BAFFLING IN VERTICAL BOILER

Stokers Available *for all* Types

Modern Stokers are able to meet requirements of any type of plant. They are highly efficient, have inherent high overload capacity and are exceedingly flexible in regard to load variations.

DESPITE THE remarkable advances made in burning coal in the pulverized form during the past ten or fifteen years, the bulk of the coal burned under the power boilers of the world is still burned on mechanical stokers of one type or another. This does not imply that stoker firing possesses inherent advantages over pulverized coal firing nor that this condition will always remain, but it does indicate that stoker development has kept pace with other firing developments. Indeed it is questionable whether the rapid advance made in stoker design and performance would have been made if the impetus provided by pulverized coal firing had been lacking. It has influenced all types of stokers especially the larger sizes.

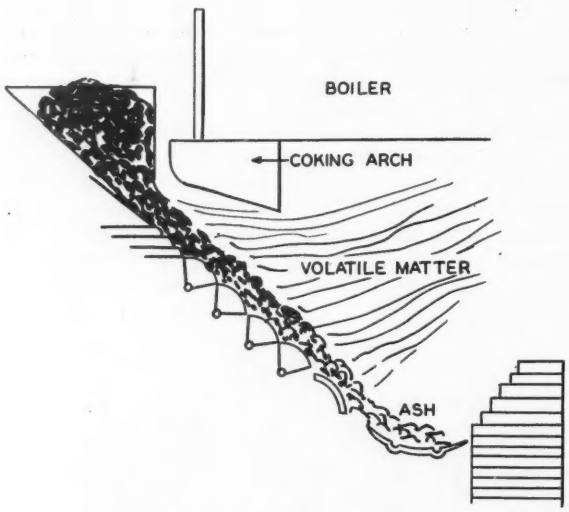


FIG. 1. DIAGRAM SHOWING THE PRINCIPLE OF THE OVERFEED STOKER

It is but natural that installations using stokers should be more numerous than pulverized fuel installations. The stoker evolved naturally and logically from hand firing; its development was inevitable. In the early years of steam engineering, boilers were small and could be fired conveniently by hand. But as the demand for boilers of larger capacity arose, hand firing became more and more impracticable, not only because of the inhuman demands it made upon the operating personnel, but because of economy and efficiency.

Today there is no excuse for any boiler, regardless of its size or application, to be fired by hand. This, today, extends even to domestic furnaces.

In its present state the mechanical stoker is a highly developed, highly refined and highly efficient apparatus and can be adapted to any condition of fuel, capacity and rating. Any kind of coal can be burned efficiently and smokelessly with some type of stoker, but no single design is equally suited to all kinds of coal. Many types, however, will handle a wide range satisfactorily. Equipped with suitable adjustments for regulating the rate of fuel feeding, thickness of bed and air supply and provided with proper combustion space they afford a highly efficient means of burning coal in mass. Naturally, in the selection of a stoker everything depends upon suiting the stoker to the coal that is to be fired on it. Not only the volatile but also the ash characteristics of the coal influence stoker selection.

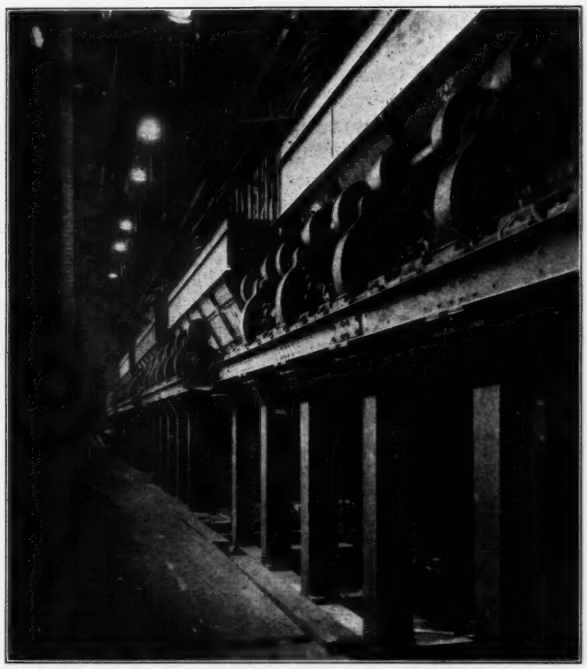
Stoker development has been materially influenced by other developments in steam generating equipment, indeed none of these developments, stokers, furnaces, boilers, pulverized fuel, forced draft, etc., should be considered alone—they must all be considered together if a true picture of the situation is to be obtained. Beginning about 20 yr. ago there was a definite movement toward increased size of steam generating equipment and increased steam output per square foot of boiler surface. This in turn brought about both increase of stoker sizes and increased sizes of furnaces. The weak point at this stage of the development was the refractory which being subjected constantly to more severe usage and which definitely set a limit of possibilities.[1]

There was doubt in the minds of many regarding the virtue of water protected furnace walls in spite of the fact that other and less perfect forms of such construction had been used for years in other places. Eventually, however, it was demonstrated that properly used water and steam cooled furnaces opened up possibilities for further improvement, increase of size and increase of capacity which could not be ignored.

Then there came the use of preheated air brought about by a change in the type of steam cycle. With pulverized coal this introduced no great difficulties but it further stimulated stoker development until today stoker manufacturers claim their stokers are perfectly satisfactory for use with air temperatures up to 550 deg. or 600 deg. F. This involved certain fundamental considerations in stoker design which demanded reconsideration of other factors. The use of highly preheated air sometimes radically changes the way in which fuel acts on the stoker. Stokers have been made both wider and longer. Increase of width may be said to increase the fuel burning capacity in direct proportion to the increase of stoker width. This is true, except for a slight difference due to the diminishing effect

[1] See Modern Stoker Equipment, by C. F. Hirshfeld and G. V. Moran, A.S.M.E. Transactions, SSP-51-50.

and Sizes *of* Boilers



Front view of the 692 sq. ft. 15 retort, 69 tuyere stokers at Hudson Avenue Station before the control panels were installed.

of side wall conditions as a per cent of the total as the stoker gets wider. Increase in length, on the other hand, produces more than a proportionate increase in fuel burning capacity.

These are some of the factors which have been involved in stoker design in recent years. They have resulted primarily in refinements, in improvements upon existing designs, no new types have been evolved. All present day stokers still fall into one of three general classifications, i.e., overfeed, chain grate or underfeed. The chain grate stoker usually is further classified as to whether it is natural draft or forced draft.

The overfeed type of stoker is adaptable only to small and medium size boilers and to steady loads. Ordinarily they operate with natural draft but sometimes this is supplemented by forced or induced draft. They are made in two forms; the front feed inclined grate type which receives fuel at the front and feeds it down an inclined grate to the ash dump at the bottom and the double inclined side feed type which feeds from both sides onto grates that are inclined downwards towards a central ash pocket.

The principle of the overfeed stoker is shown in Fig. 1. Coal fed into a hopper is delivered onto the inclined grate and made to pass under a coking arch. Air is usually admitted with the coal under the arch. The grate bars are moved in an oscillating manner so that the coal moves constantly towards the ash dump. Due to the reflection from the arch the green coal entering the furnace is submitted to a progressive distillation of volatile matter. The gases which are thus distilled off mix with the secondary air and as this mixture is drawn into the heated combustion space combustion occurs. The fixed carbon in the fuel undergoes combustion as the fuel is carried further into the furnace. Stokers of this type will use from 70 to 80 per cent excess air but they burn certain types of coal successfully.

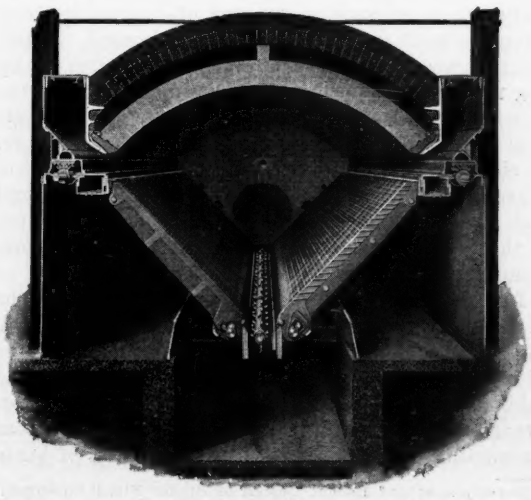


FIG. 2. DOUBLE INCLINED TYPE OF OVERFEED STOKER

Fundamentally there is no difference between the front feed and the double inclined side feed type of overfeed stoker. A typical example of this type is shown in Fig. 2. At either side of the furnace there is a coal magazine. At the bottom of the magazine are the stoker boxes which rest upon coking plates. A clinker grinder is shown at the center. An air cooled arch is sprung across the whole furnace.

Chain Grate Stokers

The chain grate stoker is very simple mechanically and in practice has proved highly successful in burning a variety of coals. It consists essentially of an endless coal conveyor which continuously receives fresh fuel at the front and carries it into the furnace where it is burned. The fuel bed remains undisturbed and the ash is discharged at the rear. Their construction varies for different makes and different types of coal but in general they are made up, a multiplicity of individual links or clips, mounted on rods or racks, the assembled unit being carried on rolls or skids and driven by sprockets.

As the fuel moves into the furnace on a chain grate stoker, the moisture is first vaporized. Next the vola-

tiles are distilled and then the fixed carbon is burned. Finally the ash is disposed of and the cleaned grate bars return underneath to the front. In many instances this type of stoker is designed for operation entirely on natural draft, but they are also designed for forced draft operation.

The natural draft chain grate stoker is most applicable to the burning of free burning, high volatile bituminous and sub-bituminous coals as well as lignite as found in the west.

The arrangement in the forced draft type is not different from the natural draft type, except that under the active length of the upper layer of the chain they are provided with a series of transverse, independent forced draft compartments, each with a separate control for regulating the air pressure under that portion of the grate.

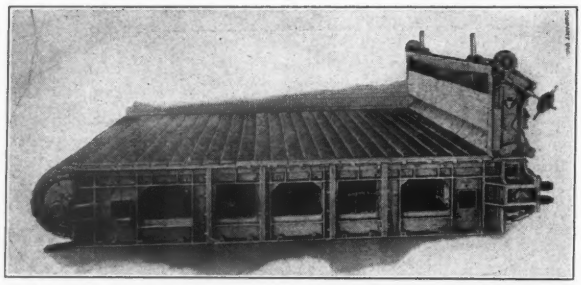


FIG. 3. TYPICAL FORCED DRAFT CHAIN GRATE STOKER

Forced draft chain grate stokers were developed in two different sections of the country for two different purposes, in the east for burning the smaller sizes of anthracite and in the middle west for bituminous coals characterized by low softening temperature of the ash, and generally by comparatively high water and ash contents and low heating value per unit of weight. Chain grate stokers are not well adapted to the burning of coals containing small amounts of ash as a certain quantity of refuse is required to protect the grate itself. This is one of the disadvantages of the conventional chain grate stoker; as the fire approaches the rear end, there is no action tending to crowd the coal together to retain a fuel bed of uniform thickness. In one type of stoker an attempt has been made to overcome this by alternating narrow moving chains with still narrower stationary grates. The addition of these stationary surfaces, constituting about 30 per cent of the entire grate area so impairs the conveying influence of the remaining 70 per cent of the moving parts that the partly burned coal must be pushed along by the good fuel back of it.

An arrangement of the forced draft chain grate used with small sizes of anthracite is shown in Fig. 4.[2] The reversed arch over the end of the stoker is a characteristic feature of this type of furnace. The construction shown in full lines was adapted after that shown in dotted lines was found to have undesirable operating characteristics and limitations. The later design gives a much wider throat with a very long rear arch and almost no front arch. It was found that the lean and rich gases mixed very satisfactorily and combustion was more nearly complete with the second arrangement.

[2] Ibid.

Underfeed Stokers

The underfeed stoker in its wide variety of designs is adaptable to a wide range of conditions and various types of fuel. Because of the large number of this type stoker installed, it may be called the universal stoker for both industrial and central station plants. There are sizes varying from those used for small domestic heating boilers to those for large central station installations of 800,000 to 1,000,000 lb. of steam per hr. In industrial practice they possess advantages in the case of application to both new and existing installations and in their comparatively high operating efficiences. Since a basement ashpit is not essential, they can be applied to almost any boiler. Built low, their installation under low set boilers permits smokeless operation and good efficiency. They are inherently smokeless so that in this particular they give excellent results.

With the underfeed stoker, the coal is delivered into the coal hopper from which reciprocating rams force it into the throat of the retort at great pressure. This causes the coal already in the retort to rise. The coal would all rise at the front of the retort if it were not for a system of distributing pushers which operate simultaneously with the primary rams. The move-

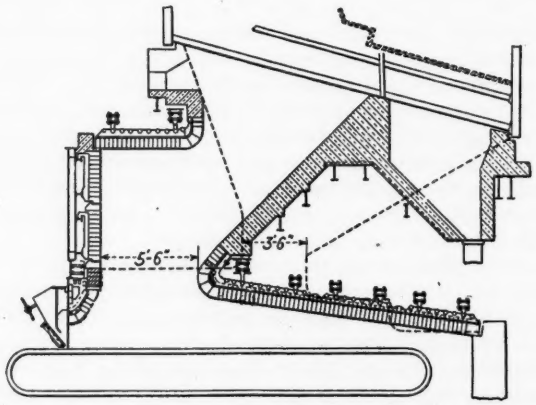


FIG. 4. ARRANGEMENT OF CHAIN GRATE STOKER USED WITH SMALL SIZES OF ANTHRACITE

ment of these pushers is so regulated in amount and speed as to cause such an amount of fuel to rise over each portion of the retort as can be burned. Alternating between the retorts are rows of tuyeres which bring the air in proper direction and quantity into the fuel where it is issuing from the retort. The arrangements vary in different makes. In some cases, the retort sides are given a reciprocating motion to effect the even distribution of coal over the grate. The fuel which is in process of coking covers both the retort and the tuyeres in a solid mass. The fuel bed is continuous from one side of the furnace to the other, as the fuel rising from the top of one retort meets the fuel rising from an adjacent retort over the tuyeres. The ash, of course, forms on the upper surface of the fuel bed. Due to the downward slope of the fuel bed combined with the breathing action by means of intermittent feeding, the ash gradually descends to the rear in increasing concentrations.[3] In the process of burn-

[3] Development and Recent Design of Stoker Fired Equipment for Steam Generation. By Jos. Worker and Jos. Bennett. A.S.M.E. Transactions FSP-50-70

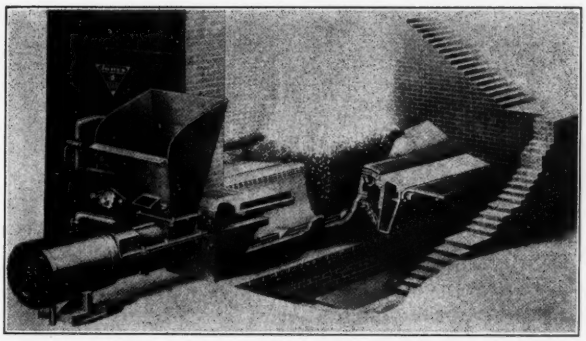


FIG. 5. SINGLE RETORT STOKER

ing, much or all of the volatile matter in the fuel is liberated over the retorts and this volatilization is retarded at the front due to the compression of the coal. There is left a devolatilized coke and ash. For the further burning of the devolatilized coke and the discharge of ash at the rear of the furnace a special form of grate is usually provided.

In this grate air is supplied to the fuel at a lower pressure than prevails in the air supply to the retorts. In some instances the grates have a horizontal stroke for moving the material on them and for burning out combustible material left in the ash. Other designs provide an agitating grate for this purpose.

The ash and clinker which forms from the ash are by this process ultimately deposited at the rear of the furnace where a receptacle or pit is provided for that purpose. The ash, clinker, and residue coke descend slowly through this deep receptacle to which small quantities of air are supplied through ventilated walls, the ash and clinker slowly coking and the combustible material slowly disappearing until finally the ash and clinker are deposited on heavy crushers or dump grates for removal from the furnace.

Underfeed stokers are made in two general forms, the single retort and the multiple retort. A single retort stoker is shown in Fig. 5. This was the earliest and simplest type of underfeed stoker to be developed. It was the start of underfeed stoking. All types of underfeed stokers today employ the principles embodied in this stoker. Coal is fed from the hopper in front of a piston and is moved forward and upward into the retort by the reciprocating action of the plunger, as already described. One or more stokers or retorts of this type may be installed in a single furnace.

The single retort stoker is used in small and moderate sized installations. Where greater coal burning capacity is needed the multiple-retort underfeed stoker is used. The multiple retort type was developed primarily for burning the better grades of bituminous coals with rather low ash content of high softening temperature.

A large unit of the multiple retort type underfeed stoker is shown in Fig. 6. This shows how feeding rams deliver charges of coal from hoppers to the entrance of the throats of the retorts. This is done under enormous pressures. The squeezing of the coal retards the distillation of volatiles until the coal passes on down in the retort. Below the rams are shown the system of distributing pushers which feed the coal into the retorts.

The combustion rates obtainable with various types and designs of stokers naturally vary over a wide range. With overfeed stokers it will usually run from 15 to 25 lb. per sq. ft. per hr. on the average. Natural draft chain grate stokers are most efficient when using 20 to 30 lb. per sq. ft. per hr. of the free burning coal for which these stokers are best suited. The maximum rates with these stokers is generally from 35 to 40 lb. and fairly good results are obtained with rates as low as 5 lb. per sq. ft. per hr. The draft required is about 1/10 in. of water for each 10 lb. of coal burned per sq. ft. per hr.

With the forced draft chain grate stoker, combustion rates are higher, running from 50 to 60 lb. per sq. ft. per hr. with bituminous coal. In some cases much higher rates have been obtained. The stoker shown in Fig. 4, using small sizes of anthracite, for instance, was operated with combustion rates as high as 75 lb. per sq. ft. per hr. To obtain these rates, however, the fuel bed must be thin and the stoker speed high (about 100 ft. per hr.) in order to maintain this rate. Ordinarily with anthracite or coke braize the average combustion rate is from 30 to 38 lb. of coal per sq. ft. of grate surface per hr. using draft pressures of from 1.5 to 2 in. of water.

In considering combustion rates with underfeed stokers, the single retort and the multiple retort types should be considered separately. The single retort type will average around 40 lb. per sq. ft. per hr., while the multiple retort stoker will run 50 to 60 lb. per sq. ft. per hr.

Naturally, regardless of the type of stoker, the combustion rates obtainable will vary with the kind of coal, the furnace design, the intensity of the forced draft, the temperature of the air. In certain cases with underfeed stokers, combustion rates as high as 89 lb. of coal per sq. ft. per hr. have been burned satisfactorily and efficiently over long periods of time. At Hudson Ave. station in Brooklyn, where 15 retort, 69 tuyere, stokers are installed, the stokers are capable of burning 57,000 lb. of coal for a period of two hours, corresponding to a combustion rate of 82 lb. per sq. ft. per hr.

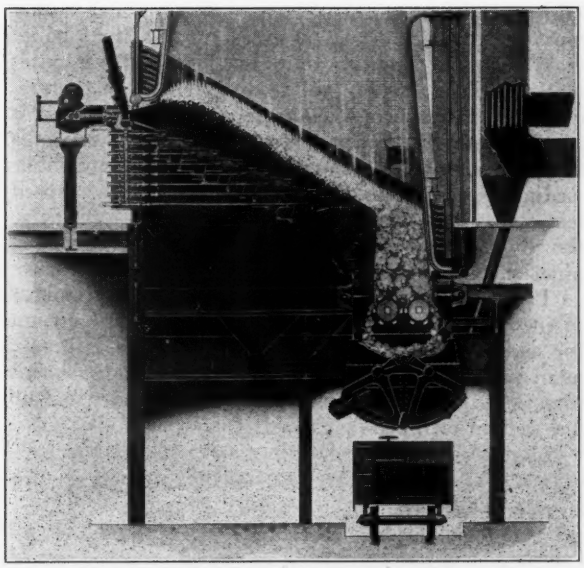


FIG. 6. MULTIPLE RETORT UNDERFEED STOKER

POWER PLANT ENGINEERING January 1934

STOKER DRIVES *show* Wide Speed Range

Stoker Drives must be adapted to the particular requirements of each installation. Various types available show wide variation in speed range. Electro-hydraulic type has range of 30 to 1.

CLOSELY RELATED to the question of stoker design is the question of stoker drive. This is a factor of great importance, which has often, in part at least, been overlooked. Many factors enter into the satisfactory control of the fuel bed on stokers but none have a greater influence than the drive controlling the rate of fuel admission to the furnace, its progressive motion on the grate and its final disposal in the ash pit. The primary movement depends upon the selection of the type of drive and the flexibility in speed control permitted by the design. The secondary movement depends upon variable adjustments made to obtain fuel bed control.

In determining the speed range required for a stoker, it is necessary to know the range over which the load on the boiler may vary. Naturally, in the selection of any stoker drive it is desirable to have as wide a speed range as possible. In Fig. 1 is shown a characteristic load curve of a central station. The range of steam output required is 6 to 1.[1]

It is apparent that a 6 to 1 range of coal feed will not be sufficient for this load curve as there are substantial variations in steam requirements due to feedwater regulation, temperature variation in feedwater supply, sudden small peaks that occur in the normal operation of the plant and peak requirements of a substantial character due to the many unusual conditions that can occur in a power plant. Moreover, to maintain a condition of this sort a reserve in fuel supply must be provided. The heat necessary to bring a boiler from a banked condition to its full load in five minutes will require a rate of coal feeding ten times that required to maintain normal rating. Furthermore, it is important to differentiate between merely a wide range of speed and a wide range of speed with any intermediate speed desired. For example, a four speed motor may give a range of 4 to 1 but it is difficult to get the right speed for a given condition, and a combination of a higher and a lower speed must be used.

It is also desirable in the case of underfeed stokers to incorporate a quick acting high speed attachment for making up any fuel bed deficiencies due to breakage or faults in operation or other considerations, this high speed attachment making it possible to speed up one section of the stoker to run faster than another.

Modern types of stoker drives are as follows:

1. Steam engine, belted or geared direct, or through line shaft.
2. Steam turbine, geared.
3. A. C. motor, direct or geared.
4. D. C. motor.
5. The electro-hydraulic drive.
6. Direct acting steam drive.

Small steam engines, usually of the single cylinder, vertical type, make very satisfactory drives. They are adaptable over a wide range of speeds and where the exhaust steam can be used they are economical.

Steam turbines also make an excellent stoker drive. They offer a wide range of speed, and while their steam rate is relatively high, if the exhaust steam can be used this is not an objectionable feature. The desirability of their use depends upon the station heat cycle or the demand for low pressure steam.

A disadvantage that is common to both the steam engine and the steam turbine is the fact that their installation is complicated by steam and exhaust piping. In this respect the electric motor has distinct advantages.

The alternating current motor is available for stoker drive in various types. The low first cost of the multiple speed (2 or 4 speed) induction motor in combination with some type of gear shifting device has favored its adoption by a large number of plants and under all ordinary conditions it is quite satisfactory. This arrangement, however, does not provide for widely changing loads. The wound rotor type of alternating current motor will improve the speed control but costs are higher and its efficiency is low at the lighter loads.

The brush shifting motor also finds application in stoker drives. An installation of this kind using automatic control is used at the Edgar station of the Edison Electric Illuminating Co. of Boston. In this installation the motor is connected to an intermediate jack shaft through a single chain and sprocket drive. A

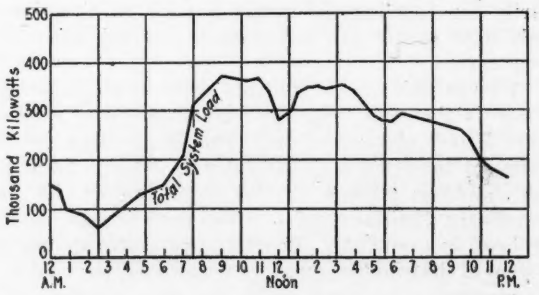


FIG. 1. TYPICAL CENTRAL STATION LOAD CURVE SHOWING A 6 TO 1 RANGE IN STEAM OUTPUT

[1]Developments and Design of Stoker Fired Equipment for Steam Generation. A.S.M.E. Transactions. By Jos. Worker and Jos. Bennett. FSP—50-70.

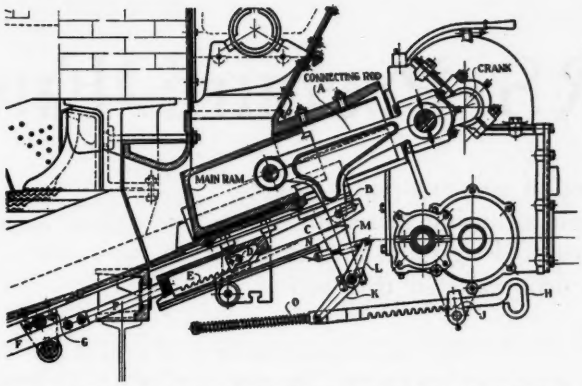


FIG. 2. STROKE ADJUSTING MECHANISM ON A MULTIPLE RETORT UNDERFEED STOKER

clutch at each end of this jack shaft allows it to drive the main stoker shaft through either of two chain drives, one with a 2 to 1 and the other with a 3 to 1 speed reduction. The automatic boiler control of the station varies the speed of the motor over its range according to the demand by shifting the motor brushes. This is accomplished by a control drive consisting of a continuously running pilot motor connected to the brush shifting mechanism through solenoid operated clutches. When the maximum stoker motor speed is reached using the lower speed drive, a yellow light indicates to the operator that the drive should be changed to the high speed.

Direct current motors are well suited to stoker drive, since they have a wide speed range and their efficiency is good over the speed range. Usually, however, converting equipment is necessary and for this reason they are quite expensive to install. Where direct current is generated for other purposes the direct current motor forms an ideal type of stoker drive.

The electro-hydraulic drive is a relatively recent development and provides all the variation in speed range that a modern stoker demands. In this system, a constant speed motor drives a pump discharging oil at a rate which can be varied to suit the load conditions of the boiler. The oil discharge is connected to a motor with a constant displacement, which of course runs at a speed determined by the quantity of oil discharged by the pump. In this way a speed variation of 20 to 1 can be obtained, or converting it into the usual terms, a boiler rating from 25 to 500 per cent can be carried. In some cases a variation of 30 to 1 has been obtained.

In many forms of underfeed stokers the direct steam drive is used. In this case the piston rod of a steam cylinder is connected directly to the coal plunger. The rate of coal feed is regulated by means of a special valve or valves having a number of possible different speeds. In some cases this valve is operated from a pulley on the forced draft fan, which is driven by an engine or turbine, the speed of which is controlled by the steam pressure. The valve controlling the coal feed is set by hand to properly proportion the coal feed to the speed of the fan. Thereafter, until a change of conditions requires resetting of the coal feed valve the speed of the fan and the rate of coal feed will increase or decrease automatically with fluctuations in the load on the boiler.

Varying Stroke of Rams

In underfeed stokers various schemes of varying the stroke of the feeding rams are used to vary the rate of coal feed. One of these is shown in Fig. 2. Here the connecting rod A between the stoker crank and the main feeding ram is made with a lower extension B, which imparts motion to the upper driving arm link C. Pawl D is rigidly fastened to the end of this link and drives the rack E, to which is fastened the secondary ram driving rod F, riding between the rollers G. The mechanism as illustrated is in the approximate positions of maximum stroke. To shorten the stroke the adjusting rod H is lifted off the tooth J and drawn backward until the desired notch of the rod is engaged by the tooth. This action draws back lower arm K and thrusts forward upper arm L of the adjusting link. Connecting link M pushes the adjusting block N forward.

Another thing that should be incorporated in the driving mechanism of a stoker, is some positive means of safeguarding the driving mechanism and the stoker from breakage in case an obstruction should block the forward movement of the coal plunger. This usually takes the form of a small shearing rod or pin which shears when the drive is subjected to abnormal strain.

On chain grate stokers this pin is usually placed directly in the main transmission. With multiple retort underfeed stokers it is often applied to each individual plunger, so that when an obstruction occurs and the pin shears, only one plunger becomes inoperative, rather than a whole series of plungers. Naturally, the replacement of this safety device should be simple and easily done. Such an arrangement is shown in Fig. 3. The safety shearing pin is shown at A, passing through the connecting rod B and the shearing ring C. Normally, the shearing ring C bears against the connecting rod yoke D. The shearing pin is the only direct connection between the connecting rod and the plunger. The forward end of the connecting rod yoke is bored out to permit the connecting rod to pass through it when the plunger becomes blocked. When the plunger becomes blocked as shown at (B) the shearing pin shears and the connecting rod moves forward without moving the plunger. On the outward movement of the connecting rod, the plunger is drawn back by means of pins in the connecting rod yoke, which slide in slots in the connecting rod on the forward movement of the rod when the plunger was blocked.

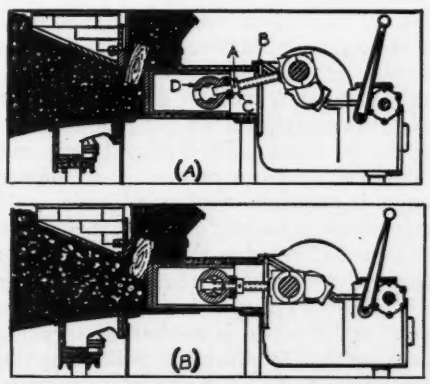


FIG. 3. SAFETY SHEARING PIN INCORPORATED IN PLUNGER MECHANISM OF AN UNDERFEED STOKER

POWER PLANT ENGINEERING January 1934

PULVERIZERS Prepare Fuel

Particles of fuel are reduced in size to a point where they float in air by equipment using attrition and impact principles. Features of design and operation of this equipment are incorporated in this article.

WHEN burning coal in pulverized form the most important piece of equipment in the preparation system is the coal pulverizer or mill. In common practice the coal comes to the mill in crushed form in sizes that will pass through 1½-in. holes. From this size, the coal size is reduced in the mill to the size best suited to the furnace, burner and character of the coal used. In general practice good results are secured with coal of a fineness that will all pass through a 50 mesh screen and 65 to 70 per cent through 200 mesh. From the pulverizer the fuel is delivered either directly to the burner or to a bin which in turn discharges to the burner.

Principles Employed by Pulverizers

Pulverizers in commercial use employ two principles in the reducing process; these are attrition, i.e., the crushing and grinding action, and impact, or reduction by hammering or pounding. Some designers claim a third principle is involved which may be designated as shearing though others would class this as attrition. All commercial mills employ both principles in their operation but in varying degrees, so that mills may be classified as either attrition or impact types, depending on which principle is used most prominently.

In the attrition type may be classed all those mills which employ balls or rollers to crush and grind the coal against a bull ring or table, or stationary and revolving plates fitted with grinding teeth that depend principally upon this action to reduce the coal to powder size. As a rule these mills depend but little on the impact action although some designs do combine the actions more than others.

Impact mills as ordinarily classified, however, do depend to a great extent on attrition to effect the final reduction of the coal size. This type of mill permits of a variety of designs which have met with success in commercial application. In Fig. 2 are shown diagrammatically the common types of impact mills. The mill shown at A receives coal at the center where it is given somewhat of a whirling motion by the arms driving the hammers, the centrifugal force and gravity carry the coal into the path of the hammers where the impact is received. Coal flows horizontally to the discharge where a whizzer allows the powdered particles to pass but returns the coarse particles to the pulverizing chamber for further reduction. The design shown by B differs from A in that the hammers travel in grooves and the coal is discharged through pockets in the casing that by means of radial ribs returns the coarse particles to be reduced further permitting only the powdered products to enter the fan chamber. A slightly different design is shown at C where the hammers travel in grooves, the product of each stage being delivered to the center of the next stage until delivered by the last stage to the exhaust fan. The design shown at D has coal delivered directly into the plane of the revolving hammers where maximum impact is received and is discharged in a vertical direction.

In all of these designs, impact plays the most important part but unquestionably attrition also is present to a considerable extent. In the design shown at E, however, both impact and attrition are employed each section performing a definite function. At F is shown a ball mill which also employs impact and attrition, the principle here being that the crushed coal and cast-iron balls are tumbled about together in the revolving pulverizing chamber, the balls pounding and grinding the coal to the powdered state when it is removed by a current of air which carries off only that fine enough to float in the air.

Although coal dryers are sometimes essential equipment in the preparation of coal for pulverizing, except when the coal is extremely moist, drying is done in the pulverizer itself by means of the air used for carrying off the powdered coal. This air may be either cold or preheated, or in some cases flue gases are used which have been found to be not only a good drying agent but also a reducer of fire hazard and an inhibiter of bin corrosion.

Classification According to Mechanical Design

Another common classification of commercial pulverizers is based on the mechanical design and is given here as it is more descriptive of the equipment than that based on principles of operation.

Classification of Commercial Pulverizers

Type	Principles Employed
Ball and Ring	Attrition
Roller and Ring	Attrition
Roller and Table	Attrition
Hammer	Impact and Attrition
Horizontal	
Vertical	
Combination	Impact and Attrition
Hammer and grinding plates	
Ball	Impact and Attrition

This ball and ring pulverizer, Fig. 3A, applies grinding pressure independently of the mill speed and the weight of the balls. It has been successfully used over the whole range of capacities demanded by present practice, and is suitable for rapidly fluctuating

For Combustion in Suspension

loads, fuels of varying grindability and service in either storage or unit firing systems. Being driven by a synchronous motor, its speed is constant but the capacity is varied by changing the rate of coal feed. Referring to the illustration it will be noted that the

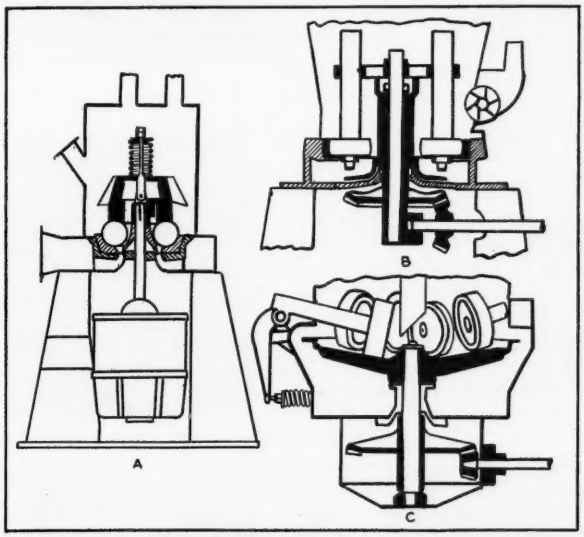


FIG. 1. CHARACTERISTIC FEATURES OF PULVERIZERS EMPLOYING ATTRITION PRINCIPLE

grinding element consists of rows of balls interposed between stationary and rotating grinding rings. The balls are propelled by the rotating ring, which floats on, and is driven by, the main shaft. The grinding pressure is maintained by springs applied to the stationary ring and adjustment can be made from outside the grinding chamber to suit the degree of fineness required. Separation of fines is accomplished by air flotation and a cone shaped separator, and when heated air is used for this purpose wet coal can be pulverized without preliminary drying.

Figure 3B illustrates a typical roller and table type pulverizer. Here rollers do their pulverizing as they revolve on stationary axes while the coal being powdered is held on a revolving dish-shaped table. Currents of hot or cold air sweep over the table and around the rollers carrying off the powdered coal by flotation. A classifying element at the top delivers the coarse particles back to the table for further grinding. The feeder employed here is of the screw type which is positive acting and gives a constant flow at a controllable rate.

One outstanding feature of the roller and ring type mill shown in Fig. 3C is the pneumatic feed control which automatically maintains the desired loading of the mill. This feature, together with the oil lubricated roller journal and development of a central lubricating system, which makes it unnecessary to stop the mill for lubrication during operation, are major improvements of recent years.

A hammer mill which is designed for low capacities is illustrated in Fig. 3D. In this type, the coal fineness is regulated by a revolving whizzer located between the grinding chamber and the fan. Important features of the mill are the swinging hammers and the tramp iron pocket which are important because of the passage of non-magnetic material, rags, wood, etc., which a magnetic separator will not remove. With larger sizes of this make of impact mill, classifiers are furnished, so designed as to give positive control of the fineness as well as positive return to the mill of the coarse particles. These mills are used in connection with the direct fired system.

Another design of hammer mill is shown in Fig. 3E which consists essentially of a cylindrical housing containing a rotor mounted on a shaft, the rotor being composed of pulverizing elements and a fan. Coal fed to the mill must be crushed to a 1½-in. size or less and does not require preliminary drying as the fan draws in sufficient air to propel the coal through the pulverizer and to support combustion. This air can be preheated when moist coal is used. As this air supply can easily be regulated, this design of pulverizer is well adapted to the unit system of firing.

The distinctive features of the hammer mill shown in Fig. 3F are, particularly, the underfeed arrangement which simplifies the feeder mechanism and tramp metal problem, the rotor revolves at a speed of 1750 r.p.m. making possible the use of a direct connected motor and giving to the swinging hammers or bars a centrifugal force that is very effective in pulverizing the coal, the exhaust fan is direct connected to the rotor shaft, one motor operates the whole unit, coarse particles are returned by a separator to pulverizing chamber for further reduction, large clearance between rotor and stationary metal parts, and a by-pass used to admit air directly to the fan when operating at low ratings, regulates the amount of coal drawn from the mill.

In Fig. 3G, the pulverizing is done by swinging hammers mounted in stages and driven at 1800 r.p.m. through a vertical shaft direct connected to a motor.

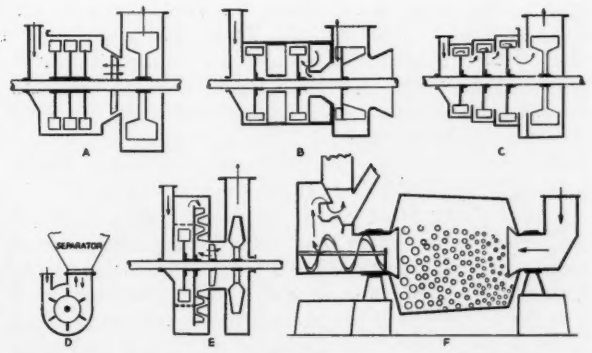


FIG. 2. FEATURES OF MILLS IN WHICH IMPACT PRINCIPLE IS PROMINENT

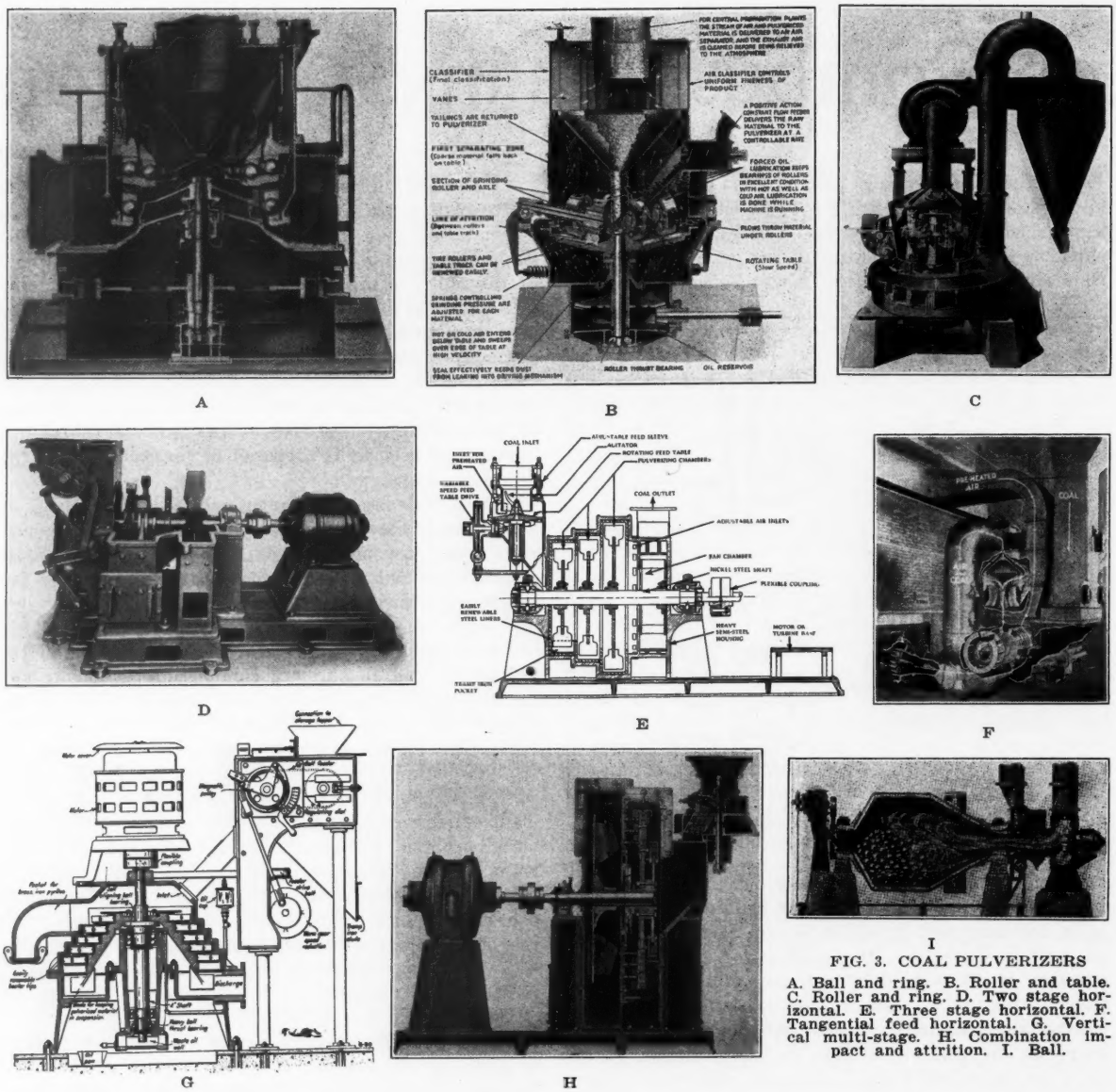

CLASSIFIER (Final classification)
VANES
TAILINGS ARE RETURNED TO PULVERIZER
FIRST SEPARATING ZONE
SECTION OF GRINDING ROLLER AND AXLE
LINE OF ATTRITION
TIRE ROLLERS AND TABLE TRACK CAN BE RENEWED EASILY
SPRING CONTROLLING GRINDING PRESSURE ARE ADJUSTED FOR EACH MATERIAL
HOT OR COLD AIR ENTERS BELOW TABLE AND SWEEPS OVER EDGE OF TABLE AT HIGH VELOCITY
SEAL EFFECTIVELY KEEPS DUST FROM LEAKING INTO DRIVING MECHANISM
AIR CLASSIFIER CONTROLS UNIFORM FINENESS OF PRODUCT
A POSITIVE ACTION CONVEYOR FEEDER DELIVERS THE RAW MATERIAL TO THE PULVERIZER AT A CONTROLLABLE RATE
FORCED OIL LUBRICATION KEEPS BEARINGS OF ROLLERS IN EXCELLENT CONDITION WITH HOT AS WELL AS COLD AIR. LUBRICATION IS DONE WHILE MACHINE IS RUNNING
PLOWS THROW MATERIAL UNDER ROLLERS
ROTATING TABLE (Slow speed)
ROLLER THRUST BEARING
OIL RESERVOIR
COAL INLET
ROTATING FEED TABLE
PULVERIZING CHAMBER
COAL OUTLET
FAN CHAMBER
NICKEL STEEL SHAFT
FLEXIBLE COUPLING
HEAVY SEMI-STEEL HOUSING
MOTOR OR TURBINE BASE
PRE-HEATED AIR
COAL
A
B
C
D
E
F
G
H
I

FIG. 3. COAL PULVERIZERS

A. Ball and ring. B. Roller and table. C. Roller and ring. D. Two stage horizontal. E. Three stage horizontal. F. Tangential feed horizontal. G. Vertical multi-stage. H. Combination impact and attrition. I. Ball.

The housing which fits over the beaters has corrugated steel rings on the inside which clear the swinging hammers or beaters by about $\frac{3}{16}$ in. Crushed coal, $1\frac{1}{2}$ in. or under, is fed from a hopper on to a belt feeder provided with a magnetic pulley which removes the tramp iron, then drops down the feed chute to the top stage of the pulverizer. Rocks and metal parts that pass the magnetic pulley are discarded into a trash pocket from the first stage, the coal passing from stage to stage as it is ground finer and finer until it is finally discharged from the last stage where it is picked up by the air stream furnished by fan blades located at the bottom of the revolving element.

One of the most distinctive combination type of pulverizer is shown in Fig. 3H, illustrating how coal to be pulverized drops directly from the feeder to the lower half of the first stage of the pulverizer, here a series of swinging hammers reduces the raw coal to a granular state. The partially powdered coal then passes around the outside of the rotor to the final stage of pulverization. Here moving pegs carried on a rotating disk travel between stationary pipe fastened to the housing, and reduce the granular particles to the proper fineness. From this compartment the pulverized coal is drawn through a series of rapidly rotating rejector arms which mechanically separate the coarse particles of coal from those of the desired fineness and throw them back among the moving and stationary pegs for further pulverization. The fan then delivers the coal in suspension with air to the burner or bin as desired. About 10 per cent of the air required for combustion is required to carry the coal from this pulverizer when the unit is working at maximum capacity.

Of the ball type of impact and attrition pulverizers, that shown in Fig. 3I is typical. The coal to be pulverized is fed into one end of this revolving conical shaped barrel where it is reduced by the iron balls that hammer and grind the coal particles until they are fine enough to be carried away by the current of air that is reversed in direction in the pulverizing chamber and thus controls the fineness of pulveriza-

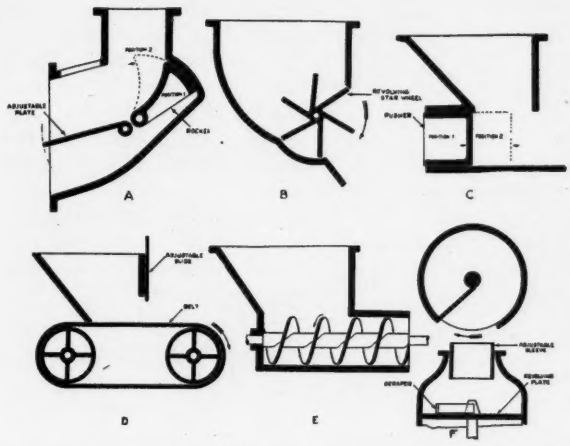


FIG. 4. PRINCIPLES OF VARIOUS TYPES OF FEEDERS

tion. When used in the direct system of firing, the amount fed the furnace is varied instantaneously by a single adjustment which regulates the raw coal feed to the mill, the amount fed to the furnace and the primary air supply. When used in the storage system, delivery is to the bin instead of the furnace.

Feeders

In a pulverized coal system, coal feeders play an important part, for upon their proper functioning rests the success of the entire installation. Specially designed feeders are essential for two distinct services in the pulverized coal system. One of these is to feed crushed coal to the pulverizer from a hopper or bin, the other is to feed pulverized coal from a bin to the burner in the storage type of system. Both of these services are exacting, requiring uninterrupted feeding under accurate control as to rate. In the design of feeders, the three principal elements to receive consideration are: (1) Surface on which the material is supported. (2) Means for removing material from supporting surface, or the feed. (3) Means for regulating the quantity removed per unit of time.

Feeders may deliver either intermittently or continuously, the former being applicable to pulverizers but are not recommended for delivering to burners while continuous feeders may be used for either service. The principles of operation of several types are illustrated in Fig. 4.

In the operation of feeders difficulties have arisen from attempting to use coal that is not uniform in size, causing irregular feeding and sometimes stoppages. Naturally, the surface characteristics of the coal affect its flowing qualities. Anthracite being hard and smooth flows readily while bituminous coal being soft and rough has a tendency to pack, and moisture is particularly bothersome. The angle of the chute leading to the feeder needs consideration as does also the material of which the chute is made. For instance, the angle of repose of anthracite is 27 deg., that of bituminous run-of-mine is 35 to 37 deg., while that of coke is 40 deg.; the minimum angle for a steel chute is given by one authority as 20 deg. for anthracite, 45 deg. for bituminous and 25 to 30 deg. for coke, but if the material is likely to contain moisture, these angles should be increased. Lignite containing small percentages of clay has a tendency, particularly when moist, to give trouble in hoppers and feeders.

With the feeder shown in Fig. 5 coal is fed to the feeder hopper from an overhead coal bunker. The cylinder of coal inside the hopper skirt D is revolved by feeder disk C and agitator A which are mounted on shaft F. This shaft is rotated from the main pulverizer shaft. Hopper skirt D is slotted to permit inward and outward movement of knife B, which adjusts the rate of coal feed. As this knife is moved inward a greater slice of coal is peeled off the revolving cylinder of coal and the amount of coal fed to pulverizer is increased. A belt from the main shaft operates counter shaft H. This shaft is connected by chains and sprockets to worm shaft I, which operates shaft F through a worm and gear. Clutch J, high speed sprocket K, and low speed sprocket L are mounted on worm shaft I so that the feeder table can be operated at either of two speeds.

In Fig. 6 is illustrated a fluffer wheel type of pulverized coal feeder commonly used to deliver the coal from the storage bin to the burner. To prevent arching of the coal in the hopper, a distributor rotates over the openings above the fluffer wheel which carries the coal around and drops it into the feeder wheel which in turn discharges into the burner pipe, thus aerating the coal while passing through the feeder and discharging in a uniformly aerated stream the coal which is metered by a revolution counter on the wheel shaft. The rate of feed is adjustable over a wide range.

The feeder shown in cross-section of Fig. 7 is designed to deliver powdered coal from a bin to the burner and makes use of a rotary plate containing two filling spouts which fill successively a series of cells placed in a fixed cell plate immediately below the filling spouts. After any cell has been filled and the spout rotated beyond that cell, a blast of air is delivered through the cell blowing the coal from the cell into a mixing chamber below the cell plate. More air is admitted to the mixing chamber and the coal is blown out of the chamber and into the pulverized fuel burner.

Control of the quantity of coal delivered is accomplished by the speed at which the filling spouts rotate. An adjustable cover plate placed between the cell plate and the filling spout plate can be moved to close half the cells. The normal speed range of the filling spouts is 4 to 1 and, since half the cells can be closed, the capacity range of the feeder is 8 to 1. Two cells are placed on each radius and alternate inside and outside

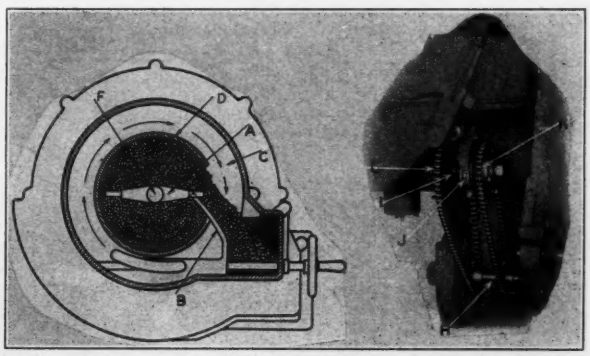


FIG. 5. DISK TYPE FEEDER FOR HANDLING PULVERIZED COAL

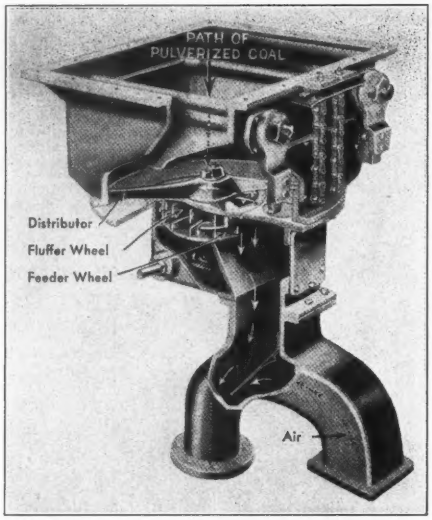


FIG. 6. DISK FEEDER WITH AERATING PRINCIPLE

cells are closed by the cell cover plate. Thus cell discharge frequency is not affected. Primary air for operating the feeder is taken from the hot air duct and is forced through the feeder by a small primary air fan. The cells are filled with coal and blown clear so frequently that coal is discharged in a continuous stream.

Maintenance of Pulverizers

In the operation of pulverizers considerable expense is attached to maintaining the pulverizing elements in condition where proper fineness can be attained and the required power per ton of output does not become excessive. In the attrition type, the grinding rings, balls and rolls are the parts which cost most to maintain, while in the impact type the paddles or hammers are the most expensive. The average life of these parts varies widely with the design of the mills and the character of the coal pulverized. The output life of balls ranges from 1500 to 10,000 tons, rolls 27,000 to 82,000 tons, grinding rings from 11,000 to 111,000 tons, paddles from 2900 to 4500 tons and fan blades from 7500 to 25,000 tons. The life of these parts, however, is being increased by the use of harder alloys which, though more expensive in first cost, are expected to be more economical due to less frequent replacements.

Just what may be expected from a pulverizing mill as to power consumption per ton of product has not as yet been very definitely determined due principally to the variable factors that enter into the problem. Among the variables that have the greatest effect upon power consumption are grindability, fineness of product, size of raw coal, moisture in coal, relative output of mill and age of pulverizing parts of mill.

Grindability is a term used to compare the ease with which coals may be reduced to a powdered state and the relative grindability is obtained by comparing the new surface to that obtained in a similar way on a coal selected as a basis for comparison. R. M. Hardgrove, who has done outstanding research work along this line, has selected coal from the Upper Kittanning seam in Somerset and Cambria Counties, Pa., as the standard against which he makes comparisons of grindability because of its easy-grinding qualities and convenience to secure. He calls its grindability 100, coals more difficult to pulverize will have a lower grindability number and vice versa; anthracite is low, in the neighborhood of 25 to 30, while such materials as petroleum coke may run as high as 175. With a grindability machine the capacity varies almost directly with grindability from 0 up to 50, above which the capacity falls off relatively so that at 100 grindability the capacity is 80.

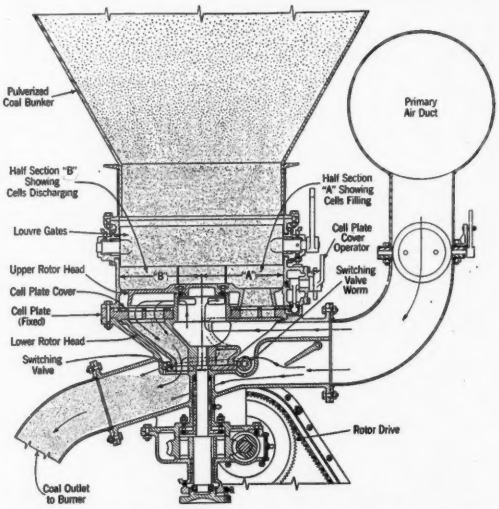


FIG. 7. CELL PLATE FEEDER APPLICABLE TO BURNERS

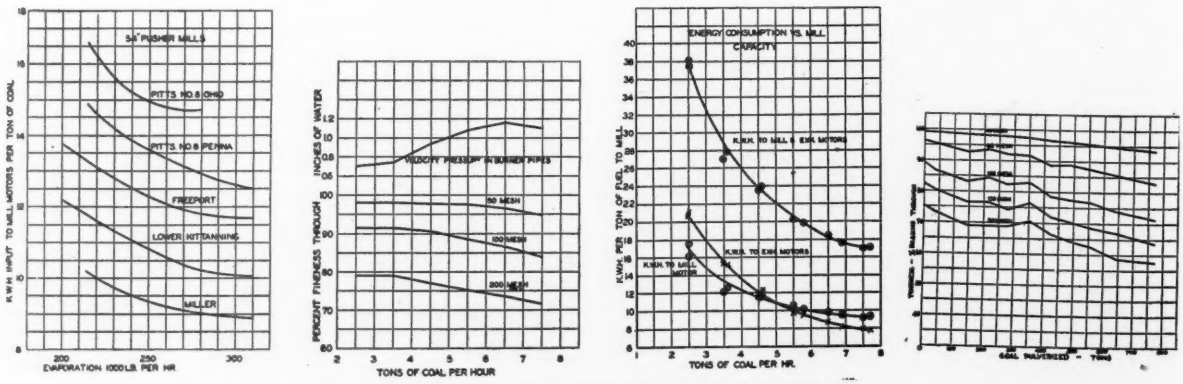


FIG. 8. TYPICAL PULVERIZER PERFORMANCE CURVES

Pulverized Coal Burners—
Atomizers of Coal Dust

Principles and equipment employed in modern practice to envelope each particle of fuel with properly conditioned and a sufficient quantity of combustion air.

SUCCESSFUL USE of pulverized coal as fuel is dependent to a large degree upon the design and operation of the burner employed. The function of the burner is to inject the fuel into the furnace so intimately mixed with air that its combustion will take place completely within the space provided in the furnace. For this reason particular attention must be paid to the method employed for mixing the fuel and air required for combustion and the manner of introducing the mixture into the furnace.

Burners first used in boiler furnaces were of the jet type with a straight circular or fan-shaped nozzle which injected the fuel into the furnace by means of a current of air just sufficient to transport the fuel through the piping. As the jet left the nozzle it ignited, combining with the oxygen in the air drawn into the furnace through vents surrounding the nozzles or advantageously located in the furnace walls. It was soon learned that this method of combustion required extremely large furnace space and a long path for the burning fuel to travel before reaching the boiler heating surface. The study of this problem has resulted in several solutions that have not only reduced the sizes of the furnace but increased the fuel capacity of the burner.

Flame Length Reduced by Turbulence

To reduce the length of the flame, more intimate and uniform mixing of the fuel and primary air has been effected, cross currents of secondary air causing turbulence are employed, forced draft is employed on secondary air and combustion air is preheated. As to the direction taken by the flame in the furnace, burners may be located in the arch, giving a vertical down and back path to the flame, an arrangement commonly used with slag tap furnaces or slow burning coal; in the front wall with the general course of the flame horizontal toward the center of the furnace; two burners in the opposite walls with flames opposing each other; or four burners located in the corners of the furnace with the flames directed horizontally at an angle to the side walls of the furnace, thus giving a circular path to the flame with the burners directed in the tangent to the circle.

The type of burner and its location is influenced by the size and shape of the furnace, the method of removing the ash, the type of furnace walls, whether water cooled, air cooled, ordinary refractory or slag-protected water walls, and the character and fineness of the fuel used.

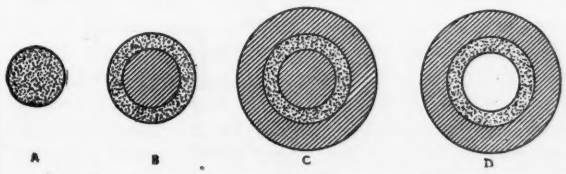


FIG. 1. DIAGRAM SHOWING POSSIBLE RELATIONS BETWEEN THE PRIMARY FUEL CARRYING AIR STREAM (STIPPLED SPACE) AND THE SECONDARY AIR STREAM (CROSS HATCHED) IN CIRCULAR BURNERS

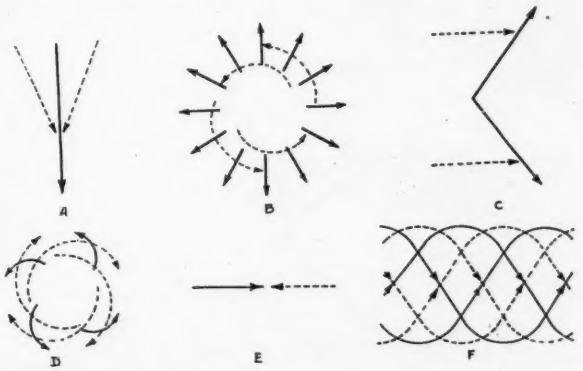


FIG. 2. TURBULENCE IN FLAME IS PRODUCED AT BURNER TIP BY IMPINGING AIR STREAMS UPON EACH OTHER AS INDICATED HERE FOR CIRCULAR BURNERS WHERE PRIMARY AIR AND FUEL STREAM IS INDICATED BY SOLID LINES AND SECONDARY AIR STREAM BY BROKEN LINES

Preheated primary air may be supplied to the pulverizer for drying purposes and thus used as carrying air for the fuel; heated secondary air may come from air cooled furnace walls or heat reclaimed from other parts of the boiler setting; or both primary and secondary air may be preheated by reclaiming heat from stack gases or exhaust or bled steam from the turbine. In any case the size of the air passages in the burner must be so proportioned as to allow the correct amount of air to pass, hot air requiring larger openings or greater pressures than cold air, consequently more power for driving the fans.

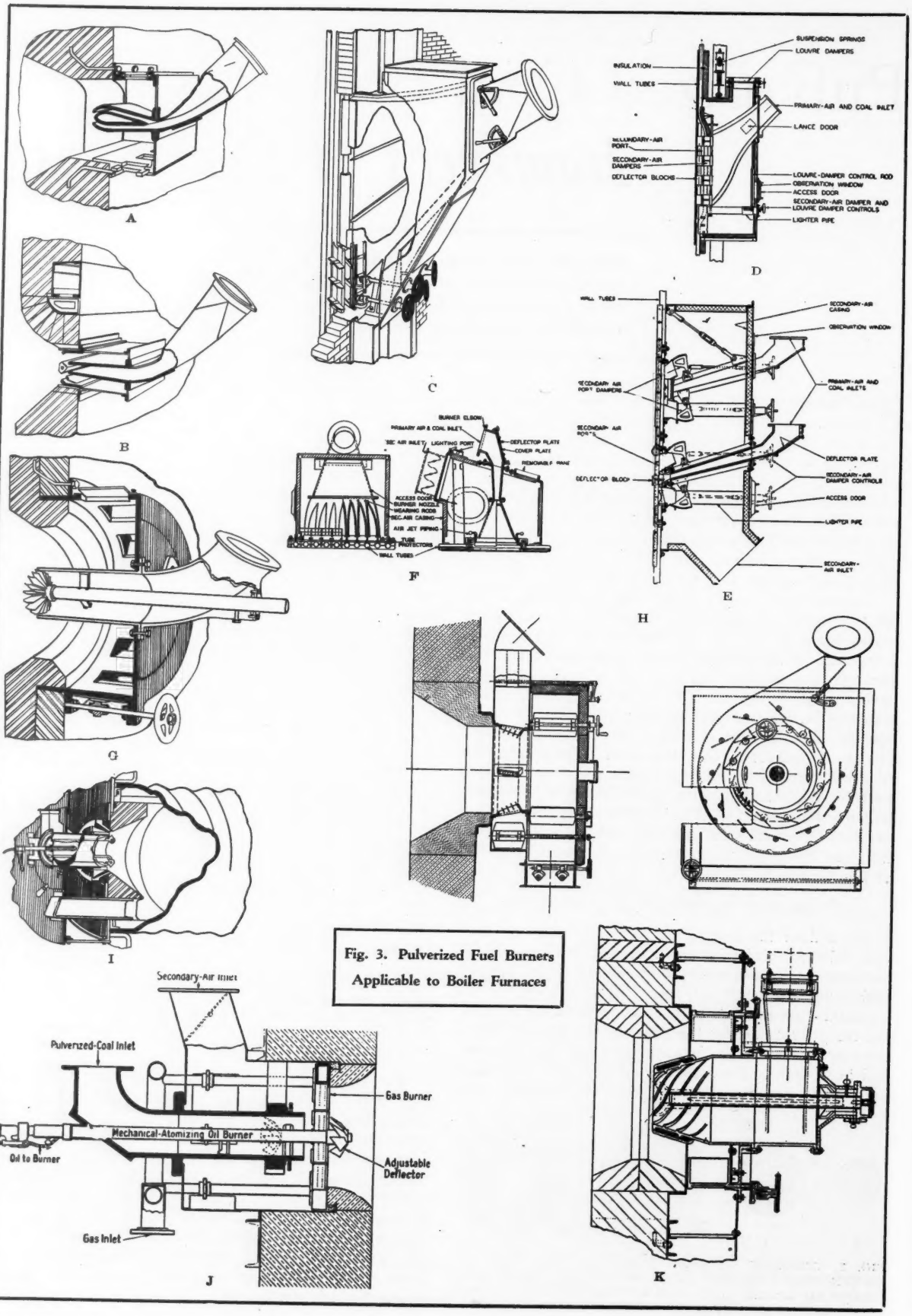


Fig. 3. Pulverized Fuel Burners Applicable to Boiler Furnaces

Primary air varies in both amount and pressure with the different methods of supplying the air direct from blowers or from fans driven with the pulverizer rotor, also with load conditions, fineness of pulverization and character of fuel. The percentage of primary air to the total required for combustion varies in practice from 10 to almost 100 per cent in the case of some unit fired systems.

Methods of Inducing Turbulence

As the coal carrying stream of air flows through the burner it may be in one of the forms of jets shown in Fig. 1. Several methods of inducing turbulence to reduce flame length are shown diagrammatically in Fig. 2.

Primary or carrying air supply is regulated according to load conditions either at the pulverizing mill in the case of the direct firing system or by means of dampers at the feeder in the case of storage systems. Secondary air is usually controlled by dampers in the ducts leading to the burner. Turbulence may be regulated by changing the angle of the deflecting vanes or the position of the deflector plate if one is used.

Features of Modern Flat Shaped Burners

Designed for natural draft and small installations, the burner shown in Fig. 3A divides the fuel bearing stream into an upper and lower channel. These streams impinge on each other as they leave the burner orifice in the stream of secondary air, resulting in a turbulent short flame suitable for horizontal firing.

The flat, horizontal burner shown in Fig. 3B delivers the primary air and coal stream against a V-shaped tip or deflector placed in front of the burner. This deflector is adjustable to and from the orifice of the burner and divides the fuel stream in two parts. Secondary air is delivered to the wind box and passes over the curved deflectors shown above and below the burner orifice. The pressure is such that the secondary air is driven into the fuel stream, resulting in turbulence at the burner tip and a short flame.

In Fig. 3C is shown a burner design which permits having all straight tubes in the water wall, the tube of which is adjustable to or from the vane or cone-shaped tube that may be made up from plates welded to the sides of the tube and at the pointed end which divides the stream of primary air and coal being delivered to the furnace. Secondary air is delivered between the adjacent tubes and is regulatable by deflectors so that the secondary air is directed through the stream of primary air and coal. The body of the burner is equipped with adjustable vanes for maintaining a uniform distribution of the fuel over the entire length of the burner orifice.

High capacity turbulent burning is secured by the use of the burner illustrated in Fig. 3D which is especially suited for use in water wall furnaces where the flame is directed horizontally. The primary air and fuel are discharged between the tubes while the secondary streams are directed across the path of the coal laden primary air in such a manner that a high degree of turbulence is obtained, resulting in rapid combustion and a short flame. These burners have heat liberating capacities up to 115,000,000 B.t.u. per burner per hr.

Figure 3E is another design suitable for water walls, where the stream of coal and primary air from each burner nozzle is broken up in passing between the tubes. Secondary air enters the furnace through ports immediately above and below the jets of primary air and coal, and at an angle to them, causing violent intermingling of this air stream with the fuel stream and a short, turbulent flame. Burners of this design are reported to have capacities ranging from 60,000,000 to 135,000,000 B.t.u. per burner per hr.

For vertical firing with burner located in a water-cooled arch, the design shown in Fig. 3F has been developed. This location of the burner directs the streams of pulverized coal and air downward toward the furnace floor and at a slight angle from the front wall. The U-shaped path of the flame provides sufficient travel of the coal particles for the completion of combustion before the gases strike the low rows of boiler tubes, reducing to a minimum the amount of unconsumed carbon passing out of the furnace with the gases. This design is well adapted to slag tap furnaces. The pressure loss through the burner is reported to be low but the turbulence sufficient to cause complete combustion within the furnace space. Burners of this design range in capacities up to 144,000,000 B.t.u. per burner per hr.

Circular Pulverized Coal Burners

Either natural or forced draft may be used with the circular burner shown in Fig. 3G. In this design, primary air and coal come to the burner through the elbow which directs the stream through the diffuser vanes that give the fuel a whirling motion into the path of the secondary air. Combustion is completed in a short turbulent flame. This burner can readily be adapted to the burning of oil in which case the oil burner is inserted through the pipe extension projecting through the elbow.

For horizontal installation and suitable for refractory furnace walls, the circular burner shown in Fig. 3H gives intense turbulence and positive mixing of coal and air as a result of the whirling circular motion of the secondary air striking the uniform multiple number of coal streams traveling in the line of the burner axis.

Coal, gas or oil may be fired in the burner shown in Fig. 3I, which was designed especially for internally fired boilers. A vortex tip is used on this burner with the primary air and coal delivered at a tangent, they pass through an annular orifice over the tip, leaving the orifice with a rotary action of relatively high velocity. The secondary air is also delivered in a rotary manner on the outside of the coal stream. The combination of the deflector and the rotary action results in intense turbulence of the flame.

For burning pulverized coal, oil and gas, the circular burner shown in Fig. 3J is extremely flexible in operation. Although installed primarily as a pulverized coal burner, the change to either liquid or gaseous fuel can be quickly made, or a combination of these fuels burned. The burner is installed in a horizontal position and the deflectors give turbulence to the flame. The oil burner used is of the mechanical atomizing type.

The circular burner shown in Fig. 3K is designed for burning pulverized coal or oil. Changing from coal to oil or vice versa merely requires the insertion or withdrawal of the oil tube with atomizing nozzle.

POWER PLANT ENGINEERING January 1934

Use *of* Oil as Fuel *is* Growing

Oil Burning Systems Require Equipment and a Technique in Their Operation Which Are Distinctive from Other Fuels. Equipment and Combusion Are Discussed Here.

PROVISIONS for transportation of oil throughout the country have made the use of oil as fuel for power plants more wide spread than in former years when limited transportation facilities and uncertainty of continuous supply made oil so expensive that this fuel was considered only in seaboard location and in districts close to the oil fields of the country. Pipe lines now completed pump oil from the oil bearing districts of the country to refineries in nearly all of the industrial centers where fuel oil is produced. From these centers a dependable supply of fuel oil is available either by truck or tank car transportation.

Many boiler furnaces are now being designed with provision for burning more than one type of fuel as it has been found that the additional initial cost is justified by the flexibility in choice of fuel as the comparative prices fluctuate with seasons and local conditions. These changing conditions make it almost imperative that power plant engineers, particularly those in industrial centers, give consideration to the problem of oil burning with the object of cutting steam generating costs.

Advantages of Oil as Fuel

Among the principal advantages of using oil as fuel in a furnace, as enumerated by Ellenwood and Hirschfeld, are as follows:

Low combustion losses, due to complete combustion with low excess air.

High efficiency.

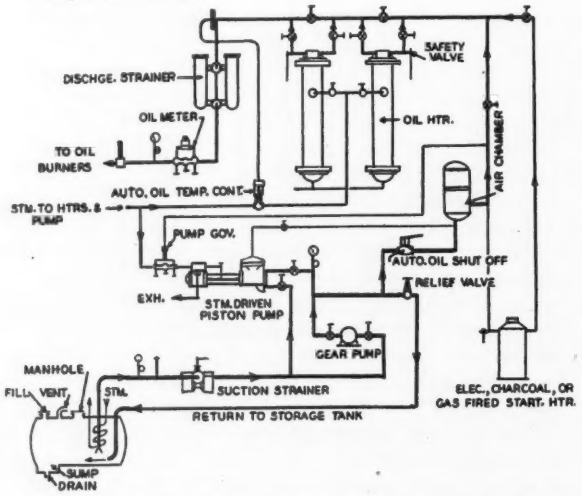


FIG. 1. ESSENTIAL EQUIPMENT AND PIPING OF AN OIL BURNING SYSTEM

Great flexibility to meet changes in loads.

Low maintenance and labor costs.

Less storage space than required with coal, due to greater heating value on either the volume or weight basis.

No deterioration of fuel during storage.

Small amount of refuse.

No banking losses.

Small draft required.

Four major problems present themselves when an oil burning system is being planned, these are: the provision for storage of the oil, proper means for conveying oil to the burners and conditioning it for atomization, provision for atomization, and supplying combustion air in such a manner as to cause complete combustion with a minimum of excess air.

Principal Equipment in Oil Burning Systems

In Fig. 1 is shown a layout of a typical fuel oil supply system for a power plant. The preferable location for the tank is beneath the ground surface outside the power house building. This location is usually chosen because of the convenience it offers for delivery of oil from tank cars or trucks and the reduction of the fire hazard. The elevation is the lowest point in the system so that the entire piping system can be drained back to the storage tank. The tank should be large enough to supply the plant with fuel during the longest probable period between deliveries and its essential features are a filling opening, a vent, a manhole, pump suction pipe, heating coil near suction pipe opening, return line pipe, drain pipe, level gage and fire extinguishing provision such as pipe for foam or gas.

Oil is drawn from the tank by means of a pump which develops a discharge pressure sufficient to give proper atomization of the oil at the burner. This varies with the design of the burner from 25 to 80 lb. for steam atomizing and 30 to 300 lb. for mechanical atomizing. A positive unfailing pressure must be maintained in the piping system so piston, plunger or gear pumps are considered most suitable. These must be controlled by pump governors that work within narrow pressure limits and the discharge pressure pulsations are smoothed out by an air chamber or an elevated reservoir.

In the line to the burner, an oil heater is used to increase the oil temperature to a point where its viscosity is best suited for atomizing in the burner em-

Practice

Boiler Fronts and Instrument Board in a Modern Oil Burning Boiler Room.

ployed. This provision makes possible the use of oils having different viscosity values at the same temperatures. The heater should have by-pass piping and be valved so that the oil may be heated or not, as the case requires; the definite viscosity at the burner being the required condition, this being governed by the temperature of the oil.

Other essentials of the piping system are strainers, preferably one in the suction lines to the pump and another in the line to the burners, an oil meter, a thermometer at the burner, a pressure gage and a pressure relief valve which prevents an over pressure on the burners and discharges excess oil back to the storage tank.

Oil Burners

For use in boiler furnaces, the devices known as burners have but two functions, that of atomizing the oil and mixing the resultant mist with combustion air as it is injected into the furnace. This atomization may be accomplished by means of a jet of steam, air or gas, or by mechanical means such as a rotating disk or a spray nozzle.

Steam jet oil burners, the general principles of operation of which are shown in Fig. 2, have been used extensively in power plants chiefly because they are low in first cost, can be installed quickly in coal burning furnaces, and firemen can learn to handle them quickly. Several inherent characteristics, however, place them at a disadvantage as compared to mechanical atomizing burners; some of these are noise, added moisture in furnace and flue passages with accompanying lowered efficiency and augmented corrosion, waste of feedwater due to steam for atomizing, and limitation of capacity per burner. The rotary disk type burner has application in small furnaces such as those used in small apartments and residence buildings but has such limited capacity that it need not be considered for installation in power plant furnaces. The mechanical atomizer type of burner employing some form of spray nozzle is now used almost extensively in power plants.

In general, engineers investigating the merits of oil burners determine their ability to atomize the oil just prior to its coming in contact with the air for combustion, to mix intimately with the oil spray, to cause complete combustion with the least amount of excess air, to produce a flame short enough for the furnace space available, to atomize without drooling, to permit of a sufficiently wide range of capacity control, and to give quick and easy access to burner parts for inspection, cleaning and repairing.

Using steam, blower air or pressure air as an atomizing agent, the design of burner shown in Fig. 3A is manufactured in capacities up to 150 gal. per hr. Fuel oil is delivered by gravity or pump to the central passage of the burner and just before it reaches the tip it is given a whirling motion by a spiral passage then discharges in a cone shaped spray being further atomized by a jet of steam or air which reaches the tip through a spiral passage having its turns in the opposite direction to those in the oil passage. Fine adjustment and regulation is accomplished by a micrometer valve in this oil supply line. The oil pressure carried averages 75 lb., steam pressure the same or slightly lower, or when air is used its pressure should be 10 lb. for light oils and 20 lb. for heavy oils.

Thorough inside mixing of oil and steam is accomplished by the burner shown in Fig. 3B before the atomized spray is discharged through the burner tip into the furnace. In this atomizing chamber of this burner, the oil is spread over a conical surface in a thin film by the action of the steam issuing from tangential holes which are located in the oil supply orifice and in line with the taper of the cone. At a point midway down the cone this whirling film is joined by steam from another series of jets which impinge angularly upon it and break the fuel up into minute particles. The entire mass is carried around the outer edge of the cone and there, reversing the flow, is caused to dash together from all sides and mix in a small chamber located at the base of the delivery pipe through which it passes to the burner nozzle or tip in an atomized condition.

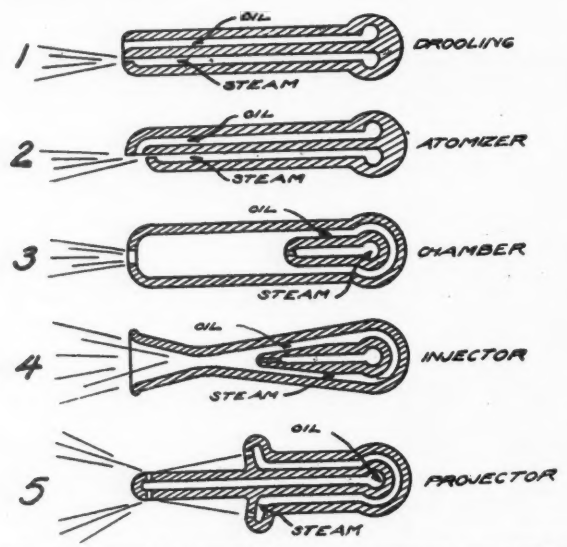


FIG. 2. TYPICAL METHODS OF ATOMIZING OIL WITH STEAM

For capacities from 15 to 175 gal. per hr., the design of burner shown in Fig. 3C is suitable for application in boiler plants. Oil enters and surrounds a main jet of high velocity steam from an expanding nozzle. Primary atomization takes place here and further on in the passage are holes that introduce tangential jets of steam that rotate the oil in the mixing chamber, whence it passes, fully atomized, through nozzles into the furnace space where it mixes with combustion air preferably introduced through a register surrounding the burner. The nozzles may be designed to produce any shape of flame, usually a hollow cone.

Figure 3D illustrates the principle of an oil burner that employs purely mechanical means of atomization. The burner proper which consists of nozzle, tube and gooseneck, fits into a holder socket that is permanently fastened to the register body or the furnace front. The gooseneck fitting contains a fine mesh strainer which catches grit, scale or other foreign particles in the oil. The burner nozzle consists of a body, tip and spiral. The fine pitch spiral gives the oil a rotating motion which causes it to discharge through the tip in a finely atomized cone-shaped spray, which permits complete combustion when mixed with the proper amount of combustion air usually introduced by natural or forced draft through a register surrounding the burner.

In Fig. 3E is shown a mechanical atomizing burner assembly which includes a strainer and an air regulating cone. The burner comprises a burner pipe and an adjusting pipe through which it extends into the air register. The atomizer assembly consists of a tip, an atomizer, an atomizer nut and a clamping nut. Atomizing is accomplished by giving the oil a whirling motion as it enters the tip for discharge into the furnace space. The capacity of the burner is determined by the size of the orifice in the tip.

With short oil passages to reduce friction and required pressure, the mechanical atomizing burner shown in Fig. 3F has capacities ranging with the size of the burner from 20 to 100 gal. per hr. corresponding to 75 to 3000 boiler hp.

Wide range operation of the mechanical pressure atomizing oil burner shown in Fig. 3G is obtained by equipping the burner with two separate atomizing chambers and tips, one of variable size and the other fixed. These tips are arranged in the end of the burner, the smaller (of fixed size, designed to supply the minimum quantity of oil practicable) in front of the larger which is of such size as required for the maximum load. They are supplied with oil through two separate and independent lines, both operating from the single supply pump, the main line, at full pump pressure and the auxiliary line at reduced pressure. This design makes possible the varying of the quantity of oil atomized by the burner over a wide range. By means of a master control valve any number of burners can be regulated simultaneously.

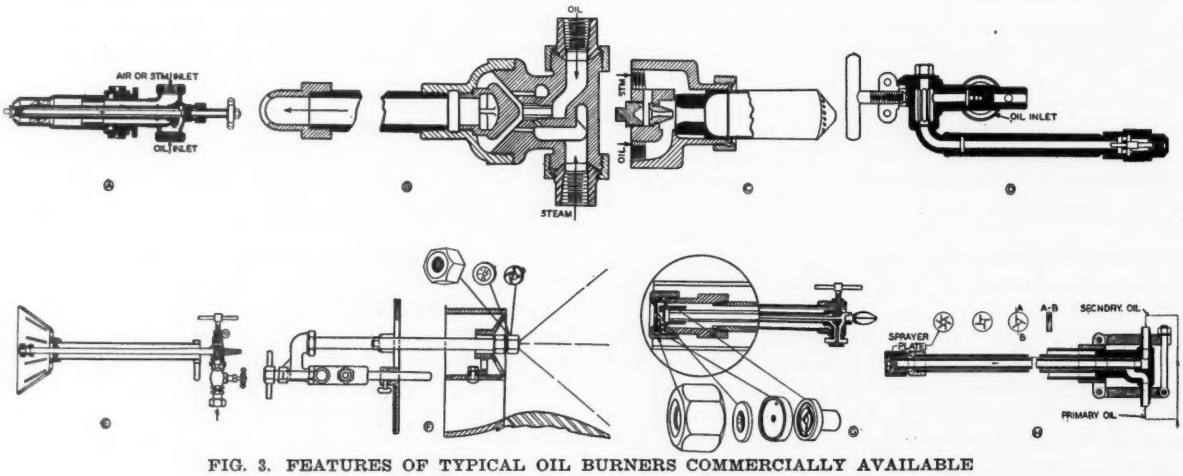


FIG. 3. FEATURES OF TYPICAL OIL BURNERS COMMERCIALLY AVAILABLE

The assembly and details shown in Fig. 3H are the essential features of another wide range mechanical atomizing oil burner. Fuel oil is delivered to the burner tip through two passages, an outer annular passage carrying the primary oil stream and the center oil barrel carrying the secondary oil. Both primary and secondary oil meet at one nozzle and a sprayer plate at the burner tip. Changes in capacity with a given nozzle and sprayer plate are secured by operating at low capacities with primary oil alone and at high capaci-

ties with primary and secondary oil together. A capacity variation of four to one with any one combination of sprayer plate and its nozzle may be secured.

Methods of Control

Two methods of controlling the amount of oil burned in boiler furnaces are recommended in modern practice. One of these is the provision of a by-pass to return a governed quantity of oil from the burner to the supply tanks or burner header; the other method employs two distinct oil feeds to the burner tips one or both of which may be used. By either of these methods minimum loads of around 25 per cent of the rated capacity may be carried without impairing the efficiency of combustion.

Admission of air to oil burning furnaces should receive careful consideration in the design of the furnace. Modern practice would indicate that combustion air should be admitted through registers immediately surrounding the burner although some installations have obtained satisfactory results by admitting additional air through ports or checker work in the bottom of the furnace. Where high rates of combustion are required either forced or induced draft or both may be necessary, for low rates, however, natural draft serves adequately. In all cases, the installation must provide means for controlling the intensity of the draft and its distribution in the furnace according to load conditions so that complete combustion with a minimum amount of air may be secured at all times.

Oil burners with their surrounding air registers are usually located in the front wall of the furnace if this location is the most convenient. Where oil is used as a supplementary fuel, however, side wall or even bridge wall location may be the best arrangement, the principal controlling factor here being that the flame must not impinge on the furnace wall. The furnace must also provide sufficient outlet for radiant heat as furnace temperatures are higher than with coal and ignition arches, necessary with certain types of stokers, are likely to undergo rapid deterioration if oil burning is employed without taking the shape of the furnace into consideration.

Heat release possible in an oil burning furnace varies considerably with the type of burner and the type of furnace walls. With both mechanical and steam atomizing burners in water wall furnaces maximum heat releases in power plant practice have reached 57,000 B.t.u. per cu. ft. of furnace space per hr.; in air cooled walls, 54,000 for mechanical and 35,000 for steam atomizing; and in solid refractory wall furnaces, 52,000 for mechanical and 32,000 for steam atomizing burners.

When it is desired to start the fire in an oil fired boiler, all draft controls should be adjusted to give air for the immediate combustion which will take place, start the pump then place a torch in front of the burner and turn on the oil. Never admit oil to a dark furnace as it may form an explosive mixture. As soon as the flame is produced regulate the air supply to give an incandescent flame which should not extend back more than 6 in. from the burner, beyond which it usually becomes transparent, runs into violet and towards the end changes to red. With too little air no violet is seen, the red darkens and smoke appears.

GAS As Boiler Fuel *is* Advancing in Use

Gas Burners are readily applied to Boiler Furnaces designed for other Fuels and its combustion characteristics make it an Excellent Fuel for Power Plants, either primary or secondary, depending upon the economics of the specific price conditions

AS A FUEL for the generation of steam in power plants, gas has many qualifications which make it close to the ideal. First cost of installing the necessary equipment is comparatively extremely low. Perfect combustion can be secured with the least amount of excess air and its accompanying losses. Control of the rate of combustion can be accomplished without complicated equipment. Labor, maintenance and repair costs are extremely low. Delivery to the plant is through pipe lines thus doing away entirely with truck or railway service for fuel transportation, and no ashes

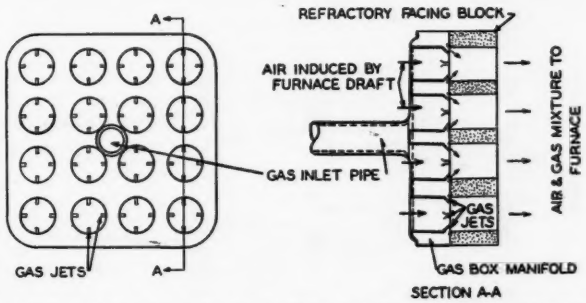


FIG. 1. ONE FORM OF JET TYPE GAS BURNER SUITABLE FOR INSTALLATION IN THE FURNACE WALL

are produced. Evaporation per square foot of boiler heating surface resulting from the use of gas as fuel averages higher than with coal and compares favorably with oil. Furnace maintenance is low, when properly designed with burners correctly located. Smoke, fly ash and objectionable stack gases do not present serious problems with gas fuel.

On the other hand, the price of gas in most communities prohibits its use although during recent years a network of gas lines has been built connecting the natural gas fields of the country with many cities in the industrial districts and the price charged is now making it a real competitor of other fuels. Natural gas is usually high in hydrogen content, in fact this element more than any other gives it its high heat value, and when burned forms considerable water vapor which promotes oxidation of the economizer and air preheating elements of the steam generating system. Other gases, such as coke oven, producer, blast furnace, water gas, oil gas and acetylene, are used in power plants but in most cases they are by-product fuels available to the plant, and their consideration is a special problem of the plant in which the gas is available. Natural gas and oil combined, however, are used to generate somewhat over 10 per cent of the electrical

output of the utility plants of the country and their possibilities should not be overlooked.

Gas Burners

Many designs of gas burners are available but, for use in a power boiler furnace, those employing the multiple jet, the bunsen or venturi tube, or an arrangement for using more than one kind of fuel are being employed almost exclusively.

Multiple jet burners have a field of application where the gas pressure is low and natural draft is depended upon to furnish combustion air. When operated at normal rating, these burners give complete and efficient combustion in a comparatively small volume, at other ratings, however, the air-gas ratio for proper combustion is difficult to maintain and the furnace volume required becomes excessive at high rates. This type of burner, therefore, does not offer much flexibility of capacity and is not best suited to power boilers subjected to variable loads. The furnace volume should be about 2.5 cu. ft. per horsepower developed with draft at 0.14 in. of water.

Venturi tube burners which employ the principle, originated by Benson, of premixing air and gas offer great flexibility of load with efficient combustion throughout a wide range of capacity and high heat

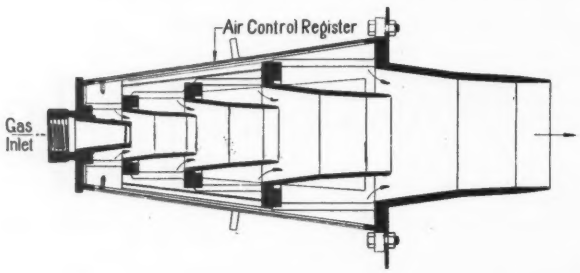


FIG. 2. VENTURI TUBE TYPE GAS BURNER, CIRCULAR NOZZLE WITH MULTIPLE AIR MIXING JETS.

release in the furnace volume. With these burners, gas pressure as well as combustion air pressure may be varied to suit load conditions. Forced, induced or natural draft may be employed and preheated air is used in some plants. The furnace volume required with this type averages 1.5 cu. ft. per horsepower developed by the boiler with draft around 0.08 in. of water. The reasons given for this differential between the jet and venturi tube type burners are that the venturi type, into which gas is injected through a nozzle, inspirates and intimately mixes a correct volume of primary air which is properly proportioned to the amount and velocity of the gas, and the inherent feature of injection of primary air makes the venturi type relatively independent of draft, and this in turn reduces the tendency to infiltration of air through leaks of the setting.

Combination burners may be designed for pulverized coal and gas; oil and gas; pulverized coal, oil and gas used separately or in any combination. Usually burners of this type are installed in plants where market price fluctuations warrant the installation of equipment that will permit a quick change from one fuel to another which is shown at the time to be the most economical.

Due to the complications necessarily entering into the design, the efficiency of combustion obtained from these burners can hardly be expected to be as high as would be obtained with venturi burners for gas only installed in furnaces also designed for gas only, but the usual design simulates the venturi type producing turbulent mixing at the throat of the burner that

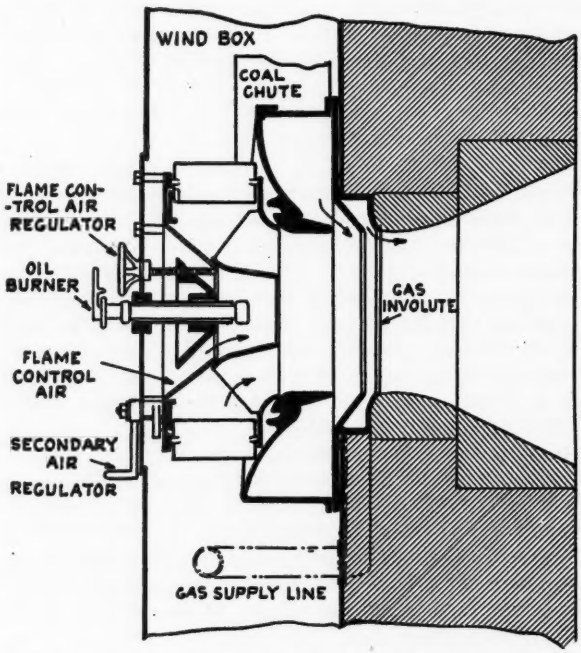


FIG. 3. COMBINATION BURNER FOR PULVERIZED COAL, OIL OR GAS

fosters rapid and complete combustion in furnaces with volumes greater than are necessary where gas alone is burned and the results obtained have been an economical benefit in many utility power plants.

Except in boilers having radiant type superheaters, steam temperatures will be higher with gas fuel than with coal or oil, so some engineers use gas in combination with pulverized coal or oil to raise the superheat temperature during peak loads when the temperature otherwise has a tendency to drop. Another practice is to use gas as fuel periodically as a means of keeping down the accumulation of soot on the heating surfaces of the boiler.

Method of Controlling Rate of Combustion

Three methods of controlling the rate of combustion are in use on gas burners serving steam boilers, these are throttling, step and off-and-on.

With the throttling method of control, a steam actuated diaphragm valve is generally used to control the firing of the boiler in accordance with the demand for steam as indicated by the variation in steam pressure. This valve controls, by its position, the gas pressure at the burner and the draft mechanism which proportions the amount of combustion air to the gas being fired. This type of control is frequently recommended for boilers whose load is above 50 per cent of its rating.

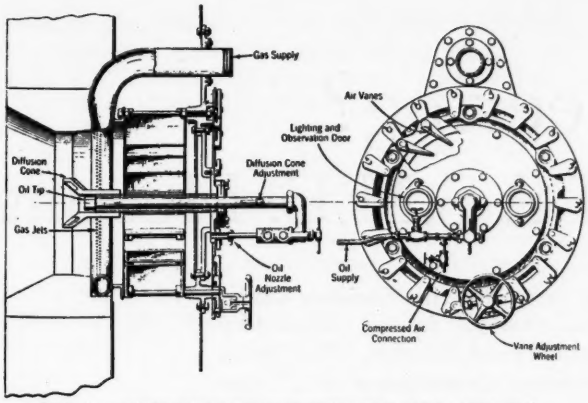


FIG 4. COMBINATION BURNER FOR GAS AND OIL

The step method differs in that adjustments take place in steps and for each step the most efficient adjustments can be made, set and locked so that it remains constant independently of the operators. Such a control is adaptable to almost any installation although with rapidly fluctuating loads considerable jumping from one step to another will take place.

Where a moderate variation of steam pressure is not very objectionable or an attendant is not continuously in the boiler room, the off-and-on method of control can be applied with satisfactory results. In this method, the burner and draft control equipment are adjusted for their rated capacity which must be considerably in excess of the normal boiler load. When the steam pressure reaches its predetermined maximum, the gas and combustion air controls are closed until the predetermined minimum steam pressure is reached when the gas and air are again turned on and combustion started by a pilot burner. This control method has been found satisfactory in many plants.

Heat release in furnaces fired with gas may be considerably higher than with coal or oil, the maximum values reported by utility companies range as high as 60,000 B.t.u. per cu. ft. in water walled and solid refractory furnaces, and 57,000 B.t.u. in air cooled furnaces. Furnace temperatures range in the neighborhood of 2000 deg. F. but this may be increased by the use of preheated combustion air, a practice known to increase boiler efficiency as much as 10 per cent. Users report, however, that an increase of 400 deg. in furnace temperature due to preheated air is uneconomical due to increased furnace maintenance.

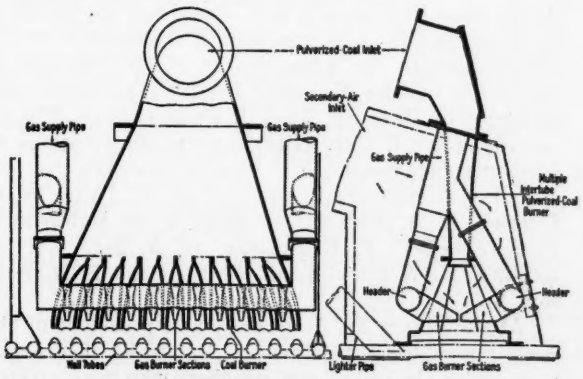


FIG. 5. MULTIPLE INTER-TUBE BURNER FOR COMBINATION FIRING OF PULVERIZED COAL AND GAS

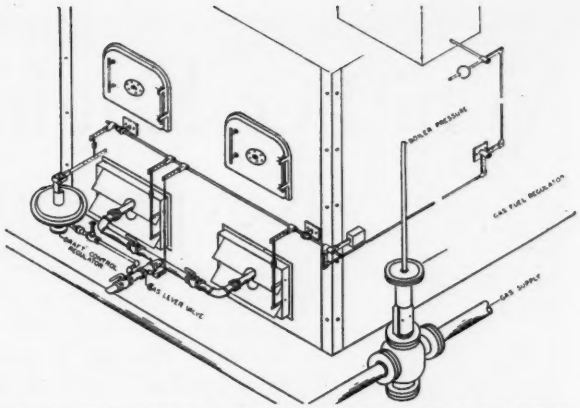


FIG. 6. DIAGRAM OF THROTTLING METHOD OF CONTROLLING GAS BURNERS

Excess air required for the combustion of gas is low compared with other fuels used in power plants, ranging in practice from 8 to 30 per cent with an average of 17. This is equivalent to a flue gas analysis range of 8 to 25 per cent CO_2 with the average of 13.

Typical efficiency and temperature curves for a boiler of large capacity are shown in Fig. 7, which show results obtained with a boiler in Long Beach Station No. 3 of the Southern California Edison Co. while burning natural gas. The heat balance efficiency reported here is obtained from the flue gas analysis and temperature while the fuel efficiency is obtained from the weighed and metered water and fuel. The difference of a little over one per cent between the two values represents the radiation and unaccounted losses.

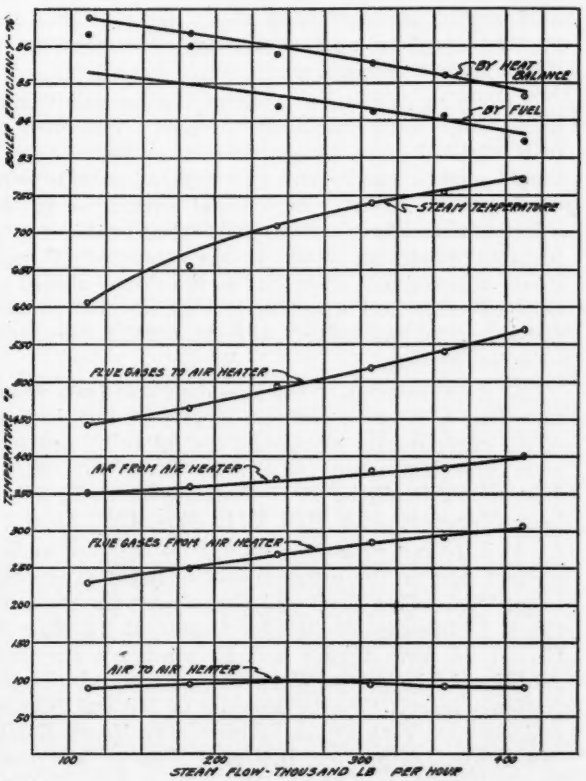


FIG. 7. TYPICAL BOILER PERFORMANCE WITH NATURAL GAS FUEL

Whither Engineering?
Keynote of A.S.M.E. Meeting

Definite Progress in Unification of Engineering Societies Under a Central National Administrative Agency Stressed by A. A. Potter, Retiring President, in Final Message. Paul Doty, Newly Installed President, Promises Continuation of Efforts.

STRESSING the need for engineers to take an interest in public welfare and solve the distribution problem as thoroughly and completely as they have the production problem, A. A. Potter, retiring president of the American Society of Mechanical Engineers, sounded the keynote of the Fifty-Fourth Annual Meeting held in New York, December 4 to 8, in his parting address, "Whither Engineering?"

Although qualified by training and accomplishment to undertake such a task, unification of effort is required. President Potter said, "Engineers must think and appear as a unified and well coördinated group; a well knit and well integrated solid body, not as a collection of individuals, each going his separate way; nor can the profession be properly represented by several distinct societies going their separate ways. This is the time for definite action to perfect the unity and solidarity of our profession and I am glad that the Founder Engineering Societies are now working on this important problem.

"Unifying the engineering profession will enable us to work more effectively on public questions of truly national scope. A central administrative agency should soon be supplemented by regional or state joint committees to assist governmental authorities in the solution of economic, social, political and other policy planning programs. Such groups composed of engineers, informal and alert, can do the things needed to help governmental agencies in dealing with public questions, and particularly with those which have technological aspects."

At the conclusion of the President's address Secretary Calvin W. Rice read the report of the election, tellers reporting the election of the following new officers: President, Paul Doty; Vice Presidents, W. L. Batt, H. L. Doolittle, E. C. Hutchinson, E. H. Whitlock; Managers, J. A. Hall, E. L. Ohle, J. M. Todd.

As in former years inspection trips occupied an important part on the program and included excursions to Hell Gate, East River and Hudson Avenue Stations, the S. S. Bremen, Rockefeller Center, R. H. Macy & Co., Hotel New Yorker and Feigenspan's Brewery. Additional trips on the Women's program included the National Biscuit Co., the Museum of Natural History and the New York Public Library. The Henry Robinson Towne Lecture was delivered Tuesday afternoon by David Cushman Coyle, the subject being "High Production and the Distribution Problem."

In addition to the President's Reception and Dance on Tuesday night, the Junior award was conferred on Townsend Tinker, Ross Heater & Manufacturing Co., for his paper, "Surface Condenser Design and Operating Characteristics." This paper in the estimation of the award committee ranked with the papers considered for the Melville Medal.

This latter medal was conferred on W. E. Caldwell, United Electric Light & Power Co., for his paper "Characteristics of Large Hell Gate Direct Fired Boiler Units." Presentation was made Wednesday evening following the Annual Dinner at Hotel Astor.

"WE may expect that their efforts will result in a central national administrative engineering agency, competent to deal with the joint activities of the engineering societies and to pass upon questions of general import without, however, interfering with the technical fields or with the autonomy of the various engineering groups. Engineers have been credited with outstanding success in organizing industrial projects of great magnitude; let us now utilize some of the ability displayed in this work to unify and coordinate our professional welfare."
From the Presidential address of A. A. Potter.

At the same time the A.S.M.E. Medal was conferred on Ambrose Swasey, past president of the Society, for his contribution to the advancement of the engineering profession and the Worcester Reed Warner Medal was conferred on Dexter S. Kimball, Dean, College of Engineering, Cornell University, for his contributions to efficient management. Fifty year badges were also presented to 13 members: C. W. Bray; F. W. Dean; W. F. Durand; H. D. Hibbard; W. Hill; W. M. McFarland; F. A. Scheffler; T. B. Stearns; G. A. Suter; E. A. Uehling; H. Webster; J. M. Whithan and F. N. Willson.

At the dinner W. A. Hanley, vice president of the Society presided as toastmaster. Calvin W. Rice, secretary, called the roll of new members who were greeted by Conrad N. Lauer, past president. This traditional

ceremony was followed by the introduction of Paul Doty, president elect, the conferring of awards and finally by the address of the evening "Capital Goods Industries," by John Dickinson, Assistant Secretary of Commerce.

The A.S.M.E. also coöperated with the Taylor Society and Stevens Institute in a celebration at the Institute commemorating the fiftieth anniversary of the graduation of Frederick W. Taylor. An exhibit of Taylor Memorabilia was available for inspection all week and on Thursday Mrs. Taylor presided at a reception and tea, which, in the evening was followed by a program.

Technical Sessions

Papers of direct interest to the power field were grouped largely in the Fuels and Water Measurement sessions on Tuesday, the Industrial, Oil and Gas Power sessions on Wednesday and the Central Station and Boiler Feedwater sessions on Thursday.

In the fuels session slag tap furnaces held the center of the stage following the presentation of Slags From Slag Tap Furnaces and Their Properties by P. Nicholls and W. T. Reid, Bureau of Mines. Seventeen stations coöperated in supplying data and usually made special runs during which samples of coal, slag and fly ash were collected.

Metallic iron in the slags was usually under 5 per cent of the total iron; in a few slags having high total iron it was 10 to 15 per cent. There was no metallic iron in any fly ash when analyzed, but there is the possibility that some may have been present when the ash left the furnace, but that it was oxidized or converted to sulphate. The combustible in the fly ash was about 6 per cent for samples collected from the gas stream; that in grab samples was high. The reduction of the iron in the fly ash was much less than in the slag.

Possibility of returning fly ash to the furnace and absorbing it in the slag was mentioned. The general feasibility of such a procedure would depend on the quantity of fly ash, the relative fusibilities of fly ash and slag, and the flow temperature of the mixture; the last could be predicted. A limited investigation was made to determine what factors were involved in the absorption of the ash by the slag.

Suggested conclusions were: Making a sludge of the ash would facilitate its general and uniform distribution over the bed and would facilitate its absorption; that some combustible is of assistance in that it reduces the Fe_2O_3, but that there is the possibility that large fly-ash additions might result in the production of excessive metallic iron; spreading the ash in a thin layer would permit more carbon to be burned and thus limit the reduction of the iron.

Results of tests returning fly ash at Hell Gate were reported by J. J. Grob, United Electric Light & Power Co. These tests were only partially successful as the fly ash had an appreciably higher fusion temperature than the original ash or slag. Injection of the dry fly ash into the furnace through a feeder caused the material to pile up near the cool walls and absorption was slow. Attempts to distribute the material caused a large percentage to recirculate through the boiler. These problems can undoubtedly be worked out however, the advantage of the slag tap furnace in reducing ash emission being unquestioned.

PAUL DOTY, PRESIDENT OF THE A.S.M.E.

Experiences along this line were also discussed by A. L. Baker, Sargent & Lundy, Inc. Some tests were cited in which no difficulty was experienced, the 24 hr. accumulation of fly ash being dumped in at once. The material lay in the form of a cone which partially melted after 6 hr. and totally disappeared after 24 hr. Difficulties with tapping, however, led to the development of a new method of continuous tapping through a hole in the bottom of the furnace. Slag drops through this hole as formed into a large tank of still water. The tank is emptied to the sluice system once a day.

This method removes some hazards of the intermittent tap system which although not serious were sometimes troublesome with certain coal, as pointed out by A. Howard, Oklahoma Gas & Elec. Co., and particularly by J. D. Donovan, City of Kansas City, Kansas, where segregation and separation of iron sulphide caused trouble by cutting the furnace floor and tapping spout and by exploding in the quenching water.

In all these problems fluidity and softening temperatures play an important part, A. W. Gauger, Pennsylvania State College, pointing out that this slag probably behaves like glass where viscosity varies with time, several hours being required to reach equilibrium. Preliminary work at Carnegie Institute of Technology also indicates a definite relation of the softening temperature with the SiO_2/Al_2O_3 ratio, in general, the softening temperatures decreasing with the ratio up as far as 2 to 2.5 after which there are indications of a slight increase. Analysis of data in the original paper showed that, in general, this ratio was greater for the slag than for the fly ash indicating either a loss of SiO_2 or a gain in Al_2O_3.

E. G. Bailey, Babcock & Wilcox Co., mentioned the flexibility of pulverized coal-slag tap bottom combinations and its independence of fuel selection. The recent practice of having the burners impinge on the slag bottom is believed to have an influence in the amount of ferric iron formed and the free tapping of slag. It is believed that these burners can be located closer than the present distance of about 16 ft. This would extend the range of satisfactory operation and reduce the cost of the furnace through higher ratios of heat liberation.

Closely related to this discussion was the paper, Burning Characteristics of Pulverized Fuels and Radiation From Their Flames, by R. A. Sherman, Battelle Memorial Institute. This paper was abstracted in an earlier issue[1] and covers investigations on four different coals. R. M. Hardgrove, Babcock & Wilcox Co., in discussing the paper, mentioned that burner characteristics would influence the results and that the preparation of the coal before burning is important. For instances, tests for Pocahontas, Pittsburgh and Hocking Valley Coal show 3, 1 and 1 per cent respectively minimum unburned carbon which agrees well with commercial tests. Data on Illinois coal showed 1.3 per cent unburned carbon, which is high compared to practice. One station in particular was mentioned where the unburned carbon in the flue dust ran approximately 0.2 per cent. This circumstance was probably due to the drying of this coal to 1.1 per cent moisture whereas it normally carries 11 to 15 per cent moisture.

Hydraulic Papers

Following the precedent established at the Spring Meeting in Chicago where the symposium on Water Hammer[2] was presented, the Hydraulic Division arranged an exhaustive program in water measurement covering known methods such as current meters pitometer, Gibson and Allen methods, their application and tests and accuracy. Eleven papers were presented, three of them being abstracted in an earlier issue[1]. Many of these papers dealt with the influence of local conditions on accuracy as observed by laboratory and commercial tests in this country and abroad.

One paper in particular created a great deal of discussion. This was Gaging for Low Head Units of High Capacity by J. M. Mousson, Safe Harbor Water Power Corp. For measuring the water used by the 42,500-hp. wheels installed at the Safe Harbor Hydroelectric Development, current meters were selected, and their use brought up numerous new problems not commonly encountered. The solution of these required much pioneer effort in applying on a large scale for the first time in America a method already in common use in Europe.

As a result of these tests an improved form of meter was secured to overcome difficulties that were encountered. In subsequent tests it was found that by using a suitable meter, measurements could actually be made in a bell-mouth section where there is considerable convergence of the stream lines. Careful comparison with the index method of measurement shows a high degree of consistency.

In discussing this paper J. F. Roberts, Power Corp. of Canada, felt that Mr. Mousson was conservative in his estimate of time saving by the vertical integration method as one day per test for computations instead of four can undoubtedly be decreased. This savings in time comes through the fact that the meters are slowly moved over the area and the mean velocity found just as accurately as though a number of set point velocity readings were taken.

L. F. Moody, Princeton University, felt that there is some doubt as to the dependability and possible limitations of the two meter angular method, a method proposed some 20 yr. ago. He felt that meters with suitable under registering and over registering characteristics in oblique flow would be developed to eliminate many present possible advantages. F. H. Rogers, Bladwin-Southwark Corp., also expressed some doubt as to the application of still water rating correction curves to field conditions.

As indicated from these and several other papers presented at these sessions there is a great mass of current meter data available in this country but the results are not well correlated. As pointed out by V. M. White, Allis Chalmers Mfg. Co., all methods of testing must be handled by someone with the necessary experience and technique if accurate results are expected. He suggested the formation of a committee to specify under what conditions the various methods would be used and what accuracy could be expected.

Diesel Engine performance as affected by details of the installation was covered in a comprehensive paper "Improving Diesel Engine Operation by the Selection of a Proper Exhaust Pipe Length," presented by K. C. Whitefield, and abstracted in a previous issue[1]. Another interesting paper presented at this session was a review of the present status of the Humphrey Gas Pump for pumping water by displacement, by F. DeP. Thomson, Sun Shipbuilding & Dry Dock Co.

This pump combines an internal combustion engine and displacement pump in a single unit, the water acting as the piston. Three installations have been made, the first two (1913 and 1930) are four cycle; the latest at Chester, Pa., is of the two cycle type. This is a 36-in. pump which has been operating over a period of 2 yr. at heads ranging from 20 to 120 ft. and capacities up to 2000 g.p.m. The two cycle design eliminates the earlier difficulties of the four cycle design which was unduly sensitive to changes in water level. It is expected that, due to the extension of natural gas facilities, this type of pump will be more popular in the future.

Liveliest of all sessions was the discussion of industrial power devoted largely to the possibilities of power interchange between industrial plants and utilities. From the point of controversy this subject promises to take the place of the old stoker vs. pulverized coal debates without, unfortunately, the advantage of close acquaintanceship of the participants which attended the latter debates. Both main papers, Coöperation Between Industrial and Public Utility Companies in the Generation of Steam and Electric Energy, by H. D. Harkins, and Some Broader Aspects of Planning for Industrial Steam and Power Supply, by V. E. Alden, were abstracted in an earlier issue.[1]

Discussion was long and spirited. Kerr Atkinson, Jackson & Moreland, mentioned the necessity for utilities to supply combined steam and power services for

[1]Power Plant Engineering, A.S.M.E. Papers at New York Meeting Reflect Technical Progress in Engineering, December, 1933, p. 532.

[2]Review of Water Hammer Theory, Power Plant Engineering, October, 1933, p. 439.

individual or group customers, the probable necessity for their purchase of old industrial boiler capacity and the importance of industrial generating capacity for peak load service. Possible utilization of diversification of industrial capacity by utilizing the public utility as a common carrier for the interchange or pooling of power was pointed out by G. B. Randall, Champion Coated Paper Co. If this is unsatisfactory at present it may be feasible to connect two or more specific industrial plants by special lines run over the right-of-way and leased from the utilities. J. H. Cather, Eastman Kodak Co., mentioned the consideration of his company and local utility of a 125-lb. mercury plant which proved not feasible at present installation prices. F. L. Bradley, Forstmann Woolen Co., reviewed conditions which led his company to build two new high pressure plants[3] to supply the demands of two mills.

IN discussing the report of the Fuels Division on Progress in Fuels Engineering, G. L. Knight, Brooklyn Edison Co., gave results of operation and tests on the eight large stoker fired units at Hudson Avenue. Top ratings of these stokers (694 sq. ft., 14,000 cu. ft. furnace volume) as installed range from 350,000 to 400,000 lb. of steam per hr. One unit was rebuilt with zoned air control and the capacity materially raised. During a 5 hr. run in November a peak output of 640,000 lb. per hr. was reached for 1 hr., 630,000 lb. per hr. for 3 hr., and 610,000 for the 5 hr. The run was terminated by the station load. A load of 640,000 lb. per hr. of steam compares to a heat liberation of about 80,000 B.t.u. per cu. ft. and about 114 lb. of coal per sq. ft. of grate area.

Much of the discussion was by utility men, C. F. Hirshfield, Detroit Edison Co., giving what might be termed the viewpoint of the open-minded and progressive utility. He has recognized the possibilities of such interchange for many years and drew attention to the fact that individual studies, not mass statistics, are necessary for detailed consideration and, more important, that there is a legal question involved in the supply of power which might possibly be sufficient in such industrial hookups to make the industrial plant operate according to the demands of the utility system rather than for its own needs.

J. A. Powell, W. S. Barstow & Co., said that detailed studies of individual projects were often not as favorable as superficial study indicated. K. M. Irwin, Philadelphia Elec. Co., thought that the great difficulty with industrial interchange is the difficulty to find plants with firm excess capacity, firm capacity being that which will, beyond a question of doubt, be available for use when needed. In considering possible plants, J. F. Muir concluded that the possibilities of exchange were limited to a few plants with large steam and small power demands. E. D. Dreyfuss, Duquesne Light Co., suggested a more plausible scheme would be for the industry to finance a plant, the utility to install the prime mover and refund the purchase price to the industrial company, title eventually passing to the utility.

Diesel engines with waste heat boilers and steam jet refrigeration with increased steam loads were led into the controversy by F. T. Leilich. Many of the difficulties of interchange were outlined by A. H. Dyckerhoff, particularly the demands made on utilities by regulatory bodies. E. C. Hutchinson, Edgemoor Iron Works, felt that the policy of utility companies to discourage installation of industrial equipment has given way to a more open minded policy. A. G. Christie mentioned the possibility of coördination of views by some kind of a regional committee under government control, as in Belgium.

Central Stations

Another session of unusual interest, perhaps because of its practical nature, was that dealing with central stations. The two papers presented at this session, "The Thermal Performance of the Detroit Turbine Using Steam at 1000 deg. F.", by F. O. Ellenwood and W. A. Carter, and "High Temperature Steam Experience at Detroit" by P. W. Thompson and R. M. Van Duzer were abstracted in an earlier issue[1].

Discussion was spirited. A. G. Christie, Johns Hopkins University, pointed out that from turbine heat rates given by the authors, station heat rates of 10,400 would be practical. This practically equals Diesel station performance and published figures on mercury vapor stations. W. E. Caldwell, United Elec. Light & Power Co., commented on the high pressure packing and also cited results of tests showing better performance with saturated rather than superheated gland sealing steam as found by the authors. As a comparison of the 8400 B.t.u. heat rate for a 100,000 kw., 1200 lb., 1000 deg. F. reheat station, W. S. Monroe, Sargent & Lundy, Inc., told of tests on a 105,000 kw., 600 lb., 725 deg. turbine which showed 9400 B.t.u. per kw.

In commenting on limiting temperatures, S. A. Moss, General Electric Co., mentioned the small exhaust gas turbine used in aeroplane superchargers. Buckets of these turbines operate bright red and he thought that this experience might be valuable when studying higher temperatures. H. E. Keeler, University of Michigan, suggested research on the specific problem of finding a better method of converting chemical to mechanical energy.

Although installations for higher than 900 deg. F. are in operation in Europe, C. R. Soderberg, Westinghouse Elec. & Mfg. Co., pointed out that the gap between these and 1000 deg. F. is large and the latter would give trouble if applied at present to large 1800 r.p.m. units. H. J. Kerr, Babcock & Wilcox Co., commented on the alloys used for the superheater tubes, mentioning that although Ka2 is still the best material to use, heat treatment has been improved so that the tubes would reach equilibrium conditions sooner and the carbide precipitation in service would be of no

[3]Forstmann Woolen Co. Builds Two Modern Power Plants, Power Plant Engineering, November, 1933, p. 464.

moment. H. Dahlstrand, Allis-Chalmers Mfg. Co., told of a chrome molybdenum steel his company is using for high temperature high pressure piping. It is not subject to carbide segregation and is welded quite readily. R. J. Brown, The Calorizing Co., gave data on heat treatment and creep tests of temperature up to 1300 deg. F. A. E. White, University of Michigan, also gave data on high temperature tests, referring particularly to the requirements of bolt stock where he said it has been quite conclusively shown that tungsten is more valuable that chromium.

Need for satisfactory boiler and superheater designs for such high temperatures was pointed out by M. D. Engle, Edison Elec. Illuminating Co., for the direct fired superheater used for the reported tests would hardly be satisfactory for more ordinary installations because of the cost. V. F. Estcourt, Pacific Gas & Elec. Co., remarked that flexibility of operation must also be considered as standby plants are sometimes required to pick up load at the rate of 1000 kw. per min.

Steam Flow

Determination of the weight of steam which may be expected to flow through a given nozzle area under varying conditions is one of the most important applications of thermodynamics, according to J. I. Yellott, Jr., University of Rochester, in his paper Supersaturated Steam. It has been shown that in the case of saturated steam the flow through nozzles often exceeds, by as much as 5 per cent, that estimated by means of the usual formula. As a solution to this problem, it has been suggested that, in a rapid expansion through a nozzle, steam might not condense when the saturated condition is reached, but might continue to expand as in the superheater region, thus becoming "supersaturated."

The research reported was undertaken for the purpose of ascertaining whether supersaturation actually takes place in the expansion of steam through turbine nozzles. It was also desired to determine the size of the droplets in which condensation occurs in flowing steam, for it was believed that the size of the droplets might have an important bearing on the subject of turbine-blade erosion. After a consideration of possible methods of attack, it was decided to use the optical method developed by Stodola, in which a beam of light is projected along the axis of a nozzle, one side of which is transparent. Because of the scattering of light by small particles, any droplets which may exist in the steam can be detected.

In discussing this paper, A. G. Christie, Johns Hopkins University (where the experimental work was carried on), said that the work is being continued to determine the effect of varying nozzle shapes and roughness of the walls. Discussion of this paper was, with few exceptions, included in the discussion of the next and closely related paper, Leaving Velocity and Exhaust Loss in Steam Turbines, by E. E. Robinson. This paper was abstracted in an earlier issue.[1]

This is a subject of prime interest to plant operators and designers and drew considerable discussion. Although agreeing that it was an excellent and thorough presentation of the subject, there appeared to be a feeling that the data and results in the paper were difficult to apply in practice. W. E. Caldwell suggested the author indicate the probable error which would be introduced for a variety of turbine types by using the leaving loss method of calculation referred to in N.E.L.A. Report 234. Sabin Crocker, Detroit Edison Co., suggested that turbine manufacturers either supply the necessary data for each particular turbine or else general data that would allow operators to calculate losses for any condition of operation. He said: "It would seem that the author could well afford to give the user's requirements further consideration in closing an otherwise commendable paper."

A. G. Christie, Johns Hopkins University, pointed out that the author indicated the different ways in which the losses would be expressed but indicated no preference. The desirability of standardizing these expressions so that all engineers would be using the same terms was emphasized. In this connection C. C. Franck, Westinghouse Elec. & Mfg. Co., felt that the most rational method of evaluating these losses would be in kw.

"THESE experiments led to the conclusion that in a rapid expansion through a simple nozzle, condensation takes place only after the steam has reached the condition approximately represented on the Mollier chart by the 3.5 per cent moisture line." *J. I. Yellott, Jr.*

Condenser performance was brought into the picture by H. G. Hubeler, Houston Lighting & Power Co., who gave data on the condenser performance at Deepwater (Tex.) Station where the circulating water varied from 50 to 95 deg. F. throughout the year. This would materially influence results as given by charts in the paper. C. R. Soderberg, Westinghouse Elec. & Mfg. Co., mentioned that moisture loss in the low pressure end of condensing turbines is another factor that should be considered as recent events indicate it is often greater than the leaving loss. P. H. Knowlton, General Electric Co., told of methods which have been used in checking the results of Mr. Robinson's work. Either models or actual turbine hoods are used, the test consisting of measurements of annulus static pressure together with the steam flow through the last stage.

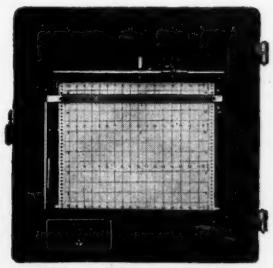
Report of Sub-Committee No. 8, Joint Research Committee on Boiler Feedwater Studies on the Determination of Dissolved Oxygen was given by C. H. Fellows, Detroit Edison Co. During the past five months further study has indicated that quantities of this gas dissolved in water may be accurately determined when in concentrations as small as 0.005 cc. per liter. Several methods of making oxygen-free water for this study have been employed, each serving to verify the correctness of the first-used more lengthy procedure.

All sessions were well attended, the registration at the meeting being approximately 2000.

New Equipment

Foxboro Pyrometer Recently Improved

ONE OF THE important features of the new recording potentiometer pyrometer designed and manufactured by the Foxboro Co. of Foxboro, Mass., is a balancing device which is so designed that it produces a large movement of the pen or print wheel for a small galvanometer deflection and moves the printing mechanism from one end of the scale to the other in a minimum of time, without requiring an extremely fast cycle or rapid motion of the mechanism. The balancing mechanism consists essentially of a V-shaped drive cam and a friction roller.



Sensing fingers, which detect galvanometer deflection, position the friction roller according to the position of the galvanometer pointer. The V-shaped drive cam then engages and rotates the roller, which in turn transmits its straight-line motion to the slide-wire contact, moving it a corresponding distance. The pen or print wheel, being mounted integrally with the slide-wire contact, moves with it, thus making the record coincide accurately with the measurement.

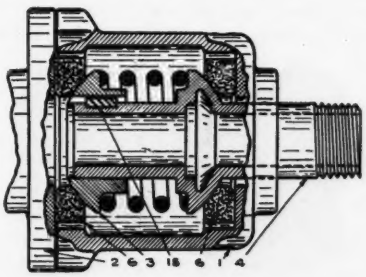
The connection compartment of the recorder, which is sealed from the interior of the case and is accessible from the outside, allows thermocouple and motor connections to be made or inspected without opening the main compartment containing the mechanism.

New General Electric Indicating Lamps

A NEW INDICATING lamp which can be used for signal-light indication or, in combination with control switches, to tell whether a circuit breaker is open, closed, or tripped automatically has been announced by the General Electric Company. This indicating lamp offers the advantages of long lamp-life and low-wattage consumption and is intended for application where good visible indication is desired. The indicating lamp unit consists of a resistor of the vitreous-enamel type, a receptacle and escutcheon constructed of insulating material, permitting their use on metal panels, a 24-v. lamp, and a color cap which may be clear, red, yellow, or green. The escutcheon, which is mounted in the panel hole from the front, and the receptacle body, which is mounted from the rear, are threaded to make a compact unit when assembled. The resistor element slides over the receptacle body from the rear, and the complete device provides for soldered connections. Binding screws are included for use when soldered connections are not desired. The color caps are made of a special translucent compound which has been treated to give more glow and greater visibility when the lamp is lighted.

Johnson Rotary Pressure Joint

FOR STEAM, gases and water, a rotary pressure joint using special carbon graphite seal rings that do not require lubrication has been developed by the Johnson Corp., Three Rivers, Mich. The pressure equalizing within the joint seals the highly polished concave surfaces of the carbon graphite seal rings against the chromium cast-iron spherical flanges. The higher the pressure becomes the tighter the

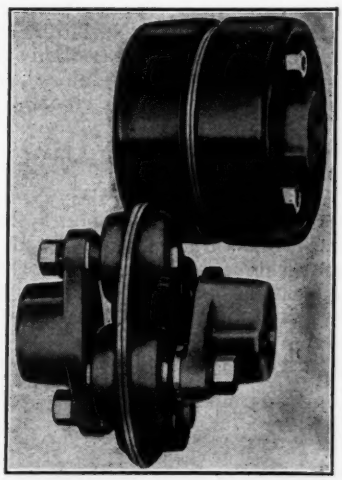


seal. The light spring merely holds the parts in place. The pressure seal feature eliminates the external adjusting spring ordinarily used to compensate for pressure changes. The lateral movement possible in addition to the angular movement allows for unusual flexibility to compensate for piping or bearing irregularities.

This joint is successfully operating on steam pressures exceeding 200 lb. and at speeds of 200 r.p.m., also on water and refrigerated brine.

Morflex Coupling

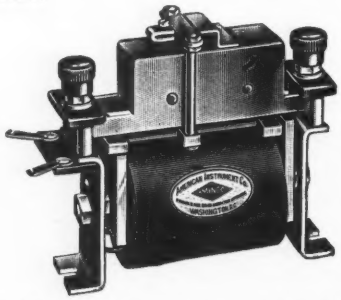
THE MORSE CHAIN CO., Ithaca, N. Y., announces a new and improved flexible coupling, known as the Morflex, using a specially developed rubber as the flexing medium. The flexing member is a complete unit comprising four moulded non-cold-flow rubber trunnion blocks spaced 90 deg. apart and set, under pressure, into a two-piece riveted steel housing. The rubber trunnion blocks are provided with steel cores or bushings also set under pressure into the blocks. The rubber is therefore always under pressure, which condition, it is claimed, prolongs its life qualities.



Two diametrically opposite blocks are bolted respectively to the armed steel driving and driven flanges which are mounted on the shafts to be connected. Thus all metal parts are rigid and all relative motion confined to the live non-cold-flow rubber. The only contact between the two shafts is through the flexible rubber bushings or blocks, and all flexing is entirely within the rubber. Shocks and uneven impulses are absorbed and sudden reversal can take place without damage.

New Supersensitive Relay

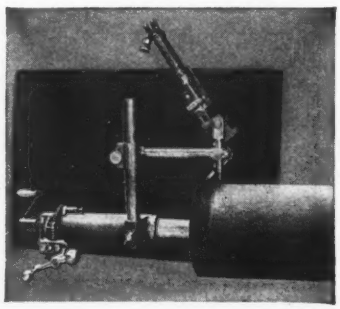
THE AMERICAN INSTRUMENT COMPANY of Washington, D. C., announces a new Supersensitive Mechanical Relay for use with thermoregulators, contact-making instruments, and other low-current devices. An operating current of 6.9 milliamperes at 6 volts d.c., or .041 watt, and a contact rating of 1100 watts, give it the remarkably high relaying ratio of 24,000:1. The high resistance of the coil of this instrument makes it ideal for use with vacuum-tube relay circuits. Its small size suggests that it will find application as component part in many types of equipment.



Overall dimensions are: Length, 3⅝ inches; height, 3½ inches; width, 1¾ inches. Coil rating: 875 ohms, 6.9 milliamperes at 6 v. Contact rating: 10 amp. at 110 v., or 5 amp. at 220 v., a.c.; available in normally open, normally closed, or single pole double throw model.

Oxweld Pipe Cutting Machine

THE LINDE AIR PRODUCTS CO., New York City, has announced a new machine known as the Oxweld pipe-cutting and beveling machine. It consists of a center rod with three spreading arms which press against the inner wall of the pipe,



holding it in position, with an arm supporting a blowpipe that can be adjusted to the desired angle of the cut. The blowpipe and arm rotate without the use of a crank for quick centering of the device, and by means of a crank when doing actual cutting.

Once centered in the pipe, the operator turns the crank causing the blowpipe to rotate evenly around the pipe, making a clean machine-like cut.

Mercury Thermoregulator

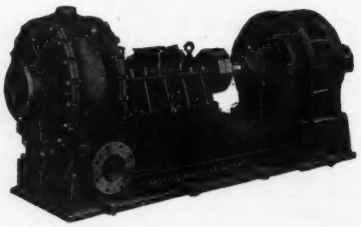
A NEW SEALED TYPE mercury thermoregulator which may be easily and accurately set at an operating temperature without resort to trial and error methods is a new product of the American Instrument Co. of Washington, D. C. To set this regulator, the bulb is immersed in liquid at the temperature at which the regulator is to be set; time is allowed for the bulb to come to temperature; and the surplus mercury is spilled off into the side reservoir by a quick tipping motion. By this simple operation, the regulator is set within 0.02° C. and is now ready to operate. The sensitivity is ±0.01° C. or better.

The setting is not affected by a 20° overshoot in temperature. Like all Aminco sealed type mercury thermoregulators, this device is intended for use with an Aminco Supersensitive Relay, or any relay that does not use more than 10 milliamperes operating current.

New Heavy Duty Dredging Pump

MORRIS MACHINE WORKS, Baldwinsville, N. Y., has added to its line of centrifugal pumps a design which has been specially developed for handling extremely abrasive mixtures against high heads.

In this new design, known as



the Type F Heavy Duty Dredging Pump, all parts subject to wear, such as the casing, impeller, disc, liners, sealing and throat rings, are made of semi-steel, manganese steel, or special Morris alloys depending on the severity of the service. The casing and impeller are of large diameter so that the pump will run at comparatively low speed even when operating against high head. The position of the impeller can be adjusted from the outside of the pump to take up wear on the suction sealing ring and prevent internal leakage. Parts subject to greatest wear are designed to permit quick and economical replacement when necessary after long service.

These pumps are built in a complete range of sizes from 4-in. to 15-in. discharge, for total heads up to 150 ft., and for operation by electric motor or belt.

Rockwood Ceiling and Vertical Drive Bases

THE NEW ROCKWOOD ceiling drive base gives to ceiling drives full range of conveniently made tension adjustment. It has special hanger arms which suspend the motor horizontally and give greater clearance from the ceiling.

The adjustable steel angles which move horizontally on the hanger arms are supported by a

ledge cast on the bottom of the hanger arms. This ledge carries the weight of the motor and takes the weight from the slotted-head screws in the arm slot.

The auxiliary pair of screws on the hanger castings serve to move the motor relative to the pivot shaft without having to support the motor weight independently while the change is being made. Once the proper belt tension has been established, the weight of the motor, lightly cradled in the belt, automatically keeps the belt at the proper tension throughout its life.

When the motor is located above the driven pulley, the springs are adjusted to support the entire motor weight plus the necessary belt tension. When the motor is located below the driven pulley the springs are adjusted to support only that portion of the motor's weight not needed to provide belt tension.

Thus the belt tension may be accurately and quickly established by the machine operator and may

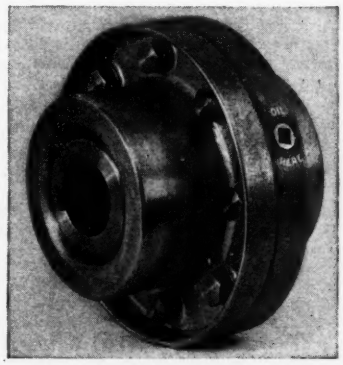
be instantly changed to accommodate different operating loads as often as desired. This adjustment is made merely by turning the two nuts at the ends of the coil-spring rods.

To simplify location of motor relative to belt length at time of installation and to provide adjustment for elongation of belt in service, the pivot shaft is movable vertically by means of convenient adjusting screws as shown.

These motor losses are products of the Rockwood Mfg. Co., Indianapolis, Ind.

New Light Duty Coupling

FAST'S COUPLINGS in small sizes and at reasonable cost for light duty are announced by The Bartlett Hayward Co., Baltimore, Md. Sizes



range from 1 to 2⅛ in. bore, horsepower from 2 to 15 per 100 r.p.m. with a maximum speed of 3600 r.p.m. The new coupling carrying the same broad guarantee as the standard design has been well tested in service over a period of three years.

A.I.E.E. Nominates Officers

THE NATIONAL NOMINATING COMMITTEE of the American Institute of Electrical Engineers, consisting of members from various parts of the country, has nominated the following official ticket of candidates for the offices becoming vacant August 1, 1934:

FOR PRESIDENT

J. Allen Johnson, Chief Electrical Engineer, Buffalo, Niagara & Eastern Power Corporation, Buffalo, N. Y.

FOR VICE-PRESIDENTS

North Eastern District:

W. H. Timbie, Professor of Electrical Engineering and Industrial Practice, Massachusetts Institute of Technology, Cambridge, Mass.

New York City District:

R. H. Tapscott, Vice-President, New York Edison Company, New York, N. Y.

Great Lakes District:

G. G. Post, Vice-President, Milwaukee Electric Railway & Light Company, Milwaukee, Wis.

South West District:

F. J. Meyer, Vice-President in Charge of Operation, Oklahoma Gas & Electric Company, Oklahoma City, Okla.

North West District:

F. O. McMillan, Research Professor of Electrical Engineering, Oregon State College, Corvallis, Ore.

FOR DIRECTORS

F. Malcolm Farmer, Vice-President and Chief Engineer, Electrical Testing Laboratories, New York, N. Y.

Nevin E. Funk, Vice-President in Charge of Engineering, Philadelphia Electric Company, Philadelphia, Pa.

H. B. Gear, Assistant to the Vice-President, Commonwealth Edison Company, Chicago, Ill.

FOR NATIONAL TREASURER

W. I. Slichter, Professor of Electrical Engineering, Columbia University, New York, N. Y.

These official candidates, together with any independent nominees that may be proposed later in the manner specified by the constitution and by-laws, will be voted upon by the membership at the coming election in the spring of 1934.

Clifford F. Messinger

CLIFFORD F. MESSINGER, President of Chain Belt Co., Milwaukee, died suddenly on the night of December 12, at his home in Milwaukee. He was in apparently excellent health, had been at the office all day on the day of his death, and was reading at home when suddenly seized with a heart attack at about 11:00 p. m., as a result of which he collapsed. Mr. Messinger was born in New Haven, Conn., September 9, 1889, and received his education in New Haven grade schools, the Boardman School of New Haven and was graduated from Sheffield Scientific School of Yale University in 1911. He entered service of Chain Belt Co. in the same year as a college apprentice since which time he progressed steadily until elected to the presidency of the company in December, 1930. At the time of his death he was a director of the Chain Belt Co., Interstate Drop Forge Co., Sivyer Steel Casting Co., and Federal Malleable Co., all of Milwaukee, and was president and director of the Stearns Conveyor Co. of Cleveland, Ohio, trustee of the Northwestern Mutual Life Insurance Co. of Milwaukee, president of the Milwaukee Metal Trades Association, and a member of the Finance Committee of the Milwaukee Community Fund.

A. Eugene Brady

ARTHUR EUGENE BRADY, Vice President and Treasurer of Jenkins Bros. died Saturday, November 4, at his home in New York City at the age of 58. Mr. Brady had been in poor health for several years, but had, during this time, maintained an active interest in the tasks of his office.

Born at Athens, Greene County, New York, September 27, 1875, Mr. Brady moved to New York City in 1893, and became associated with Jenkins Bros. the same year. Last June 1, Mr. Brady celebrated the fourtieth anniversary of this association.

Acknowledgement

WE WISH to extend our appreciation to the following manufacturers for their willing coöperation in supplying data and illustrations used in the preparation of this issue:

Air Preheater Corp.
Allis-Chalmers Mfg. Co.
American Arch Co., Inc.
American Engineering Co.
American Pulverizer Co.
Babcock & Wilcox Co.
Bernitz Furnace Appliance Co.
Bethlehem Steel Co.
Bigelow Co.
Wm. Bros Boiler & Mfg. Co.
Buffalo Forge Co.
Centrifix Corp.
Cochrane Corp.
Cokal Pulverzone Corp.
Combustion Engineering Co., Inc.
Denver Fire Clay Co.
Direct Separator Co.
Engineer Co.
Erie City Iron Works.
Foster Wheeler Corp.
Grindle Fuel Equipment Co.
Hagan Corp.
Hardinge Co.
A. J. Herschmann (Vitkovice Works)
International Engineering Works
Johns-Manville Co.
E. Keeler Co.
Kennedy-Van Saun Mfg. & Eng. Corp.
Kewanee Boiler Corp.
McClave-Brooks Co.
McLeod & Henry Co.
Natural Gas Equipment Co.
Riley Stoker Corp.
Schutte & Koerting Co.
Springfield Boiler Co.
Strong-Scott Mfg. Co.
Superheater Co.
Surface Combustion Corp.
Todd Combustion Equipment, Inc.
Henry Vogt Machine Co.
Webster Engineering Co.
Whiting Corp.
L. J. Wing Mfg. Co.

News From the Field

THE FIRM OF SHELDON, Morse, Hutchins and Easton, marketing counsellors, with offices in the Graybar Building, New York City, has been changed from a partnership into a corporation. At the meeting of the directors held recently, the following officers were elected: President, William H. Easton; Vice-president, L. W. Hutchins; Treasurer, H. A. Morse; Secretary, A. E. Welty.

HIRAM WALKER & SONS, Inc., through their engineers, Smith, Hinchman & Grylls, have placed an order with The Neckar Co., Inc., for boiler feedwater treatment equipment for their new distillery at Peoria, Ill. The system will have a capacity to treat 150,000 lb. of raw water per hour.

THE JOHNSON CORP., Three Rivers, Mich., has recently appointed the following representatives for their Instant steam water heaters and Gast air separators and aftercoolers: Advance Engineering Co., 69 Dey street, New York, N. Y.; Barrett-Christie Co., 108-112 N. Clinton street, Chicago, Ill.; Brogan & Co., 810 Race street, Philadelphia, Pa.

THE KELPO FREE-WHEELING or one-way industrial clutch has been acquired by the Morse Chain Co., Division of Borg-Warner Corporation. This clutch has been a product of the Kelpo Clutch Co., of Rockford, Ill. In the future, the Kelpo clutch will be manufactured at the Ithaca plant of the Morse Chain Co. and will be sold by Morse Chain Co. representatives. Fred M. Potgieter, formerly president of Kelpo Clutch Co., has joined the Morse organization.

THE NATIONAL ASSOCIATION of Fan Manufacturers announces the opening of an office at 5-208 General Motors Building, Detroit, Mich., for the purpose of coordinating their activities under the National Recovery Act. The Association, which has been in existence continuously since 1918, represents through its personnel the major portion of the so-called Fan and Blower Industry. The new office will be in charge of L. O. Monroe.

COMBUSTION ENGINEERING Co., Inc., New York City, announces the formation of a Marine Department and the appointment of Commander Horace T. Dyer as manager. This department will take over the work previously handled by the Marine Division of Combustion Engineering's subsidiary the Hedges-Walsh-Weidner Co. and will provide a complete service to the marine field in connection with the company's sectional header, bent tube and other types of boilers and related equipment.

THE EDISON MEDAL for 1933 has been awarded by the American Institute of Electrical Engineers to Dr. Arthur E. Kennelly, "for meritorious achievements in electrical science, electrical engineering, and the electrical arts as exemplified by his contributions to the theory of electrical transmission and to the development of international electrical standards."

This medal was founded by associates and friends of Thomas A. Edison, and is awarded annually for "meritorious achievement in electrical science, electrical engineering, or the electrical arts" by a committee consisting of twenty-four members of the American Institute of Electrical Engineers.

HEADQUARTERS of the American Society for Testing Materials have been moved from the Engineers' Club Building, 1315 Spruce Street, Philadelphia, to the Atlantic Building, 260 S. Broad Street in the same city. The increase in total floor area is some 40 per cent, while the general office space is nearly 60 per cent greater. An adequate Reception Room and Members' Lounge will be provided and friends of A.S.T.M. are cordially invited to inspect the new headquarters.

GEORGE W. BACH, for many years active in the management of the Union Iron Works, Erie, Pa., becomes General Manager of the American Sterilizer Co., of the same city, on January 1, 1934. He is still retained as a member of the Board of Directors and will serve the Union Iron Works in an advisory capacity. E. H. Brevillier, Secretary of the Union Iron Works, has been appointed General Manager to succeed Mr. Bach.

HARRY W. MCQUAID has joined the metallurgical staff of Republic Steel Corp., according to announcement made by Earl C. Smith, Chief Metallurgist. Mr. McQuaid, who is an authority on carburizing steels and case-hardening methods, is a pioneer in grain size control and collaborated in the development of the McQuaid-Ehn test which bears his name. He will devote his time with Republic to research and development work. Other changes incident to the broad metallurgical program under way at Republic include the transfer of Howard W. Burkett from Youngstown to the post of metallurgical engineer of the Buffalo district; the appointment of Elmer Larned in a similar capacity in the Chicago district and the acquisition of Harold T. Blair, metallurgical engineer, who will specialize in tin plate products. Karl Kautz, ceramic engineer, also has joined the Republic organization, and will specialize in research and field service on enameling sheets. M. J. R. Morris and E. R. Johnson will continue in their respective metallurgical capacities in the Central Alloy district, where all of Republic's stainless steel and much of the alloy steel are produced.

AT THE ANNUAL MEETING of the Board of Directors of the Exhibitors Committee, just held, the following officers were elected to serve in 1934: President: E. J. Billings, The Babcock & Wilcox Co., New York City; vice-president: L. W. Shugg, General Electric Co., Schenectady, N. Y.; Secretary: F. C. Kerns, The Texas Co., New York City; Treasurer: J. P. Ferguson, Reading-Pratt & Cady Co., Inc., Bridgeport, Conn.; Executive Secretary: W. A. Hemming, Exhibitors Committee, New York City.

THE ERIE CITY IRON WORKS, Erie, Pa., announces the appointment of Frank G. Brinig as General Manager. Mr. Brinig has been with our organization more than 30 yr., having served in various capacities in manufacturing, engineering and sales.

For the Engineer's Library

PROGRESS IN THE REMOVAL of sulphur compounds from waste gases, by H. F. Johnstone, has been issued by the Engineering Experiment Station of the University of Illinois, Urbana, Ill., as Reprint No. 2. Copies may be obtained at twenty cents each, by addressing the Engineering Experiment Station, Urbana, Illinois.

The purpose of the investigation was to determine the effectiveness and the mechanical and economic feasibility of various chemical processes in the removal of sulphur compounds from waste gases.

INDUSTRIAL HEAT TRANSFER. By Alfred Schack, translated from the German by Hans Goldschmidt and Everett P. Partridge. Published by Jno. Wiley & Sons, Inc., 440 Fourth Ave., New York. Cloth, 6½ x 9½ in., 357 pp. Price, $5.

Heat transfer, because of the many complications, is a difficult subject to cover from a practical standpoint. The present volume prepared primarily from the standpoint of the practical engineer, should be a valuable contribution to the literature of this field. The author's intention was to make it possible for the user to calculate his problem readily without spending half the time digging out the necessary data and equations. Final equations are in simple forms so that the slide rule may be used for calculating, with the addition, perhaps of a graphical chart.

Derivations are printed in small type so that they may be skipped by those not interested. The translators have carried this one step further and have marked the more important equations by a star. Equations which have a restricted use or are of empirical derivation were marked with a footnote with a reference to more complete theoretical equations. Equations and constants have been transferred from the metric to the English system of units and insofar as possible the nomenclature conforms with that of the American Standards Association.

BASIC GERMAN for Science Students by M. L. Barker. 164 pages. Size 6x9 in. Cloth. Published by W. Heffler & Sons, Ltd., Cambridge, England.

Aimed to help the science student acquire an understanding of the grammar and vocabulary necessary to a reading knowledge of articles and books on his special subject. Arranged in two parts. From the first line the student actually reads German; passages from the Bible are presented with parallel English translation to insure an immediate grasp of the meaning. The minimum essentials of grammar and a basic vocabulary make up the balance of Part I. Part II contains short extracts in German on a wide range of scientific subjects, together with an English translation on the opposite page, and a continued vocabulary of basic words and derivatives. An additional vocabulary appears at the end of the book. A compact and practical textbook that should be a valuable aid to the student interested in obtaining an intelligent grasp or reading knowledge of German.

Power Plant Construction News

Ala., Tuscumbia—Common Council will have plans drawn at once for new power transmission line to Government hydroelectric power plant at Muscle Shoals, with power substation and local electrical distributing system. Fund of $130,000 has been secured by Federal grant and loan. Lide & Adler, Woodward Building, Birmingham, Ala., are consulting engineers.

Ark., Stamps—Town Council plans installation of pumping machinery and accessories, and 50,000-gal.. capacity elevated steel tank and tower, in connection with waterworks extensions. Fund of $42,000 has been arranged for project, including pipe lines.

Calif., Crescent City—Board of City Trustees plans construction of new municipal electric light and power plant, with municipal pumping station for water service. Entire project will cost about $300,000. Financing is being arranged.

Del., Wilmington — Board of Trustees, Brandywine Sanitorium, Faulkland, Wilmington, plans new steam power house for central heating service at local institution. Cost about $50,000 with equipment. Financing is being arranged.

Idaho, Boise — Board of Public Works plans installation of pumping machinery and auxiliary equipment, pipe lines, etc., for new municipal water system. Fund of $1,296,854 is being arranged for project. C. C. Stevenson, city engineer, in charge.

Ill., Chicago—Sanitary District of Chicago, 910 South Michigan Avenue, plans large pumping plant in connection with West Side intercepting sewer project, with blower station and other mechanical units. Entire project will cost $33,948,000, and Federal aid in that amount is being secured. Engineering department in charge.

Ind., Fortville — Town Council has plans nearing completion for new municipal electric light and power plant to cost about $65,000, with equipment. Proposed to begin work at early date. Allen & Vagtborg, Inc., 205 West Wacker Drive, Chicago, Ill., is consulting engineer.

Ind., Hammond—Hammond Distilleries, Inc., plans installation of electric power and mechanical equipment in connection with plant extensions and improvements. Entire project reported to cost over $150,000. Company has arranged for stock issue totaling $1,033,000, portion of fund to be used for purpose noted.

Kan., Anthony—Common Council is arranging for Federal grant and loan for $60,000, fund to be used for new municipal electric light and power plant. Black & Veatch, Mutual Building, Kansas City, Mo., are consulting engineers.

Kan., Augusta — Common Council plans extensions and improvements in municipal electric light and power plant, with installation of new oil engine generator unit, and auxiliary equipment. M. H. Blaine, Augusta, is engineer.

Kan., McPherson — City Council is completing plans for new municipal electric light and power plant, using Diesel engine-generator units. Cost about $200,000, with equipment. Financing through Federal aid will soon be arranged. Burns & McDonnell Engineering Co., 107 West Linwood Boulevard, Kansas City, Mo., is consulting engineer.

Ky., Mount Sterling—Cleveland Windsor Co., Cleveland, Ohio, care of John W. Little, 323 Plymouth Building, Cleveland, architect, recently organized, plans one-story power house in connection with new distillery at Mount Sterling, where property has been acquired. Entire project will cost about $100,000. Wallace W. Leibner, 323 Plymouth Building, Cleveland, is engineer.

La., Many—Town Council is planning new municipal electric light and power plant to cost about $75,000, with equipment. Application is being made for Federal loan to carry out project.

Mich., Mackinac Island—City Council is planning new municipal electric light and power plant, to cost about $150,000, with equipment. Proposed to conclude financing at early date. Eugene Carey, Iron River, Mich., is consulting engineer.

Minn., Glenwood — City Council is arranging for Federal grant and loan of $140,000, for new municipal electric light and power plant, for which plans will soon be completed by Burlingame & Hitchcock, Sexton Building, Minneapolis, Minn., consulting engineers.

Neb., Hastings — Common Council has made application for Federal grant and loan for $157,912, considerable portion of fund to be used for extensions and improvements in municipal electric light and power plant, with installation of new boiler and accessories, ash-handling equipment, etc. Black & Veatch, Mutual Building, Kansas City, Mo., are consulting engineers.

N. J., Perth Amboy—Department of Public Works, Dr. John V. Smith, director, plans installation of two main pumping plants for new sewage disposal works on Raritan River, with mechanical equipment for treatment plant, chlorinator division, tanks, etc. Entire project will cost $859,000. Financing has been arranged through Federal aid. Louis P. Booz is city engineer.

N. D., Crosby—Common Council is arranging fund of $85,000, for new municipal electric light and power plant. Work will include local electrical distributing system. Plans will soon be drawn.

Ohio, Fairfield—Constructing Quartermaster, Patterson Field, Fairfield, plans early construction of new steam power plant for central heating service. Appropriation of about $200,000 will be available. A. M. Kinney, Inc., Carew Tower, Cincinnati, Ohio, is consulting engineer.

Ohio, Fostoria—Great Northern Distilleries, Inc., care of Don C. Hanover, Fostoria, president, recently organized, plans one-story power house at proposed new distilling plant on site acquired on Nickel Plate Railroad. Electric power equipment will be installed for distillery operation. Entire program will cost about $400,000. Claude H. Stone, Fostoria, is vice-president and general manager.

Ohio, Wapakoneta — Department of Public Service, George Anderegg, director, plans early construction of new municipal electric light and power plant. Cost $186,000, with equipment. Application has been made for Federal loan and grant.

Pa., Cambridge Springs—White Petroleum & Refining Co., Faber White, Cambridge Springs, head, recently organized, plans installation of power equipment, tanks, pumping machinery and other mechanical equipment in proposed new local oil refinery. Entire project will cost about $90,000.

Pa., Philadelphia—Philadelphia Steam Co., Seventeenth and Sansom Streets, has secured financing for $175,000, fund to be used for expansion and improvements, including distributing system extensions.

S. C., Columbia—Columbia Railway & Navigation Co., Columbia, T. C. Williams, president, plans construction of hydroelectric generating plant on Cooper River, near Pinopolis, S. C., with ultimate capacity of about 200,000 hp., in connection with project to make portion of Santee and Cooper Rivers navigable. Application has been made for Federal loan of $34,000,000 for development, which will include a transmission line to number of places, power substations and switching stations. Engineering department is in charge.

S. C., Greenwood — Public Works Commission, Joel S. Brady, chairman, will receive bids until Jan. 11 for pumping machinery and auxiliary equipment, filtration equipment for plant of 1,500,000 gals. per day capacity, pipe lines, etc., for municipal water system. Fund of $200,000 has been secured for program. Wiedeman & Singleton, Inc., Candler Building, Atlanta, Ga., is consulting engineer.

Tenn., Knoxville—Municipal Power Committee of City Council, W. A Cockrum, chairman, has made application for Federal grant and loan of $3,225,000, to finance construction of transmission line from city to new hydroelectric generating plant to be built on Clinch River, about 20 miles from city, by Tennessee Valley Authority, with new power substation and electric distributing system. W. W. Mynatt is city manager.

Texas, Austin — Board of Public Works, J. G. Morgan, city manager, plans extensions and improvements in municipal power plant and waterworks station, including installation of new 7500-kw. turbo-generator unit with accessories, two 30-ton cranes and other equipment. Fund of $400,000 has been secured through Federal loan. Proposed to begin work at early date.

Wash., Vancouver—Board of Port Commissioners, C. L. McKinley, president, is planning new municipal cold storage and refrigerating plant on waterfront, two-story, 180x320 ft., to cost over $100,000, with equipment. Proposed to arrange Federal loan in near future. Charles Deako, 814 West Seventeeth Street, is engineer.

Wis., Racine—Klinkert Brewing Co., care of Herbst & Kuenzli, 1249 North Franklin Place, Milwaukee, Wis., architects, recently organized, plans installation of electric power equipment in new four-story brewing plant at Racine, to be developed for initial capacity of 100,000 bbls. per annum. A boiler plant will be installed. Entire project will cost about $400,000. Architects noted are in charge.

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